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A study of the structural stability of an unbalanced sandwich composite configuration

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NAVAL POSTGRADUATE SCHOOL
Monterey, California

THESIS

A STUDY OF THE STRUCTURAL STABILITY OF AN UNBALANCED SANDWICH COMPOSITE CONFIGURATION

by

Mary Catherine Murphy

June 1993

Thesis Advisor: Dr. Young W. Kwon

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An unbalanced, sandwich composite structure consisting of TI 6AL-4V and GRP (Glass Reinforced Plastic) skins with a phenolic honeycomb core is being considered for construction of a surface ship mast which will enclose critical shipboard equipment. Stability of the structure is one of the major concerns in the design process. This research focuses on analytical and experimental studies of an unbalanced composite sandwich beam subject to a compressive axial load. The limit load of each skin material separately, and two failure modes of the sandwich construction (general buckling and core shearing), are measured in the laboratory. An analytical model is developed for predicting the limit load of th. unbalanced, sandwich composite configuration.

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A Study of the Structural Stability of an Unbalanced Sandwich Composite Configuration

by

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Lieutenant, United States Navy
B.A., Temple University 1983

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ABSTRACT

An unbalanced, sandwich composite structure consisting of TI 6AL-4V and GRP (Glass Reinforced Plastic) skins with a phenolic honeycomb core is being considered for construction of a surface ship mast which will enclose critical shipboard equipment. Stability of the structure is one of the major concerns in the design process. This research focuses on analytical and experimental studies of an unbalanced composite sandwich beam subject to a compressive axial load. The limit load of each skin material separately, and two failure modes of the sandwich construction (general buckling and core shearing), are measured in the laboratory. An analytical model is developed for predicting the limit load of the unbalanced, sandwich composite configuration.

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ACKNOWLEDGEMENTS

To Our Lord and Savior, Jesus Christ, and to My Mother.

I would like to acknowledge the financial and material support of the Naval Surface Warfare Center, and in particular, the assistance and encouragement of Dr. Vince Castelli. My sincere appreciation to Dr. Young Kwon, my thesis advisor, for his leadership and enthusiasm in the development of this thesis.

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I. INTRODUCTION

Composites are a special class of materials which have many structural advantages over monolith materials. They can be manufactured to produce desired strength to weight ratios, electrical, and acoustical properties. Composite sandwich constructions typically consist of two stiff skins separated by a lightweight core. A stiff and lightweight structure is produced in this way. The Navy has used composite materials since at least 1946 when two 28 foot personnel boats were designed using laminated plastic. Today, composites are used widely in the fleet and include items such as submarine hatches and rudders, MK 46 torpedo propellers, minesweeper hunters, and patrol craft [Ref. 1].

The purpose of this study is the support of the Navy's research and development of an Advanced Performance Mast System (APMS) for the DD963 Spruance class surface ship. The APMS project, headed by the Naval Surface Warfare Center (NSWC), Carderock Division, was initiated to develop a surface ship mast structure composed of composite materials, which encloses radars and their supporting equipment.

The present composite configuration under investigation by the APMS research team is an unbalanced, sandwich construction consisting of Titanium 6Al-4V and glass reinforced plastic (GRP) skins and phenolic honeycomb or foam core. Because the

mast supports heavy sensors and equipment, the stability of the structure is a major concern in the APMS design process. The objective of this research is to contribute an understanding of the nature of the instability behavior of the unbalanced, sandwich composite structure being considered for the APMS panels.

Initially, optimum configurations are recommended from a range of skin thicknesses and core compositions, using analytical methods developed from Whitney [Ref 2]. Shear, thermal, and hygrothermal effects have been omitted in all developments. Additionally, adhesives and any contributions by them to stiffness matrices have been neglected. Next, experimental studies are performed on skin materials and sandwich constructions provided by the Naval Surface Warfare Center. In predicting the limit loads for the Titanium 6-4 skin matẽrials, Euler's widely known formula for column buckling may be used. In his work, Vinson developed an equation for the critical buckling load of a simply, supported laminated beam, which can be employed to predict the GRP skin instability load with success [Ref. 3].

Sandwich columns subject to an edgewise compressive load may fail due to wrinkling of the faces, shear crimping near the ends, dimpling of the facings into the cells of the honeycomb core, or overall general buckling. If both the total sandwich structure and the core are very resistant to buckling and the faces too thin, wrinkling of the faces may

result. The core may fail in transverse shear when the face materials are very stiff. The effect of the shearing deformation on a deflected sandwich composite construction is found to have a significant effect on the limit load. The homogeneous equations of Timoshenko and Gere are modified to predict the failure load of the unbalanced, sandwich panels tested [Ref. 4].

The task of measuring the limit failure load is accomplished in the laboratory by use of an experimental test fixture, designed in collaboration with Dr. Vincent Castelli and Mr. Paul Coffin of the Naval Surface Warfare Center, and Dr. Young Kwon of the Naval Postgraduate School. The design achieves ideal simply supported end conditions. The experimental and predicted failure loads of the two skin materials are correlated to validate the experimental setup. The buckling instability of the unbalanced, sandwich composite configuration is then studied, and an analytical model is developed to predict the limit loads. This work, it is hoped, will be of great utility to the APMS Research and Development team at NSWC.

II. INITIAL OPTIMIZATION

This study was conducted to calculate the critical buckling stability for unbalanced, sandwich composite structures consisting of Ti 6Al-4V and GRP skins with phenolic Honeycomb or Syntactic Foam cores. In order to recommend an optimum configuration within the permissible range of thicknesses, designated by NSWC, nine unbalanced sandwich structures were considered.

The Ti6-4 skin thickness was varied from .05 to 0.2 inches while the cores were held at 1.5" and the GRP skin at 0.2". Cores were varied from 1.0 to 4.0 inches while the skins for Ti6-4 and GRP were fixed at 0.15" and 0.2" respectively. While the cores measured 1.5" and the Ti6-4 skin 0.15", the GRP skin was varied from 0.1 to 0.4 inches.

Table 1 lists the Material Properties utilized in calculation of critical buckling load.

TABLE 1. MATERIAL PROPERTIES UTILIZED FOR INITIAL OPTIMIZATION.

	Young's Modulus, E (psi)	Poisson's ratio	Density (lbs/ft^3)
Skins Ti6Al-4V[1]	16.5×10^6	.342	.160
GRP[2]	$E0^\circ 3.0 \times 10^6$ $E90^\circ = 3.0 \times 10^6$	.15	191 (estimate)
Cores Honeycomb[3] 1	23.0×10^3 compressive	0 (estimate)	3.0
Honeycomb[3] 2	100×10^3 compressive	0 (estimate)	8.0
Foam[4]	320×10^3	.35	6.0 (estimate)

[1] Metals Handbook, Vol. 2, 10th edition, for typical wrought alloy, annealed.
[2] T. Juska, Naval Surface Warfare Center.
[3] HEXCEL HFT/ Data Sheet 3200.
[4] Syntac 350C.

If a beam configuration is assumed, i.e., v = 0, then the governing equations for displacement of a simply supported plate subject to a uniaxial compressive load reduce to [Ref. 2]:

$$A_{11}\frac{\partial^2 u}{\partial x^2} - B_{11}\frac{\partial^3 w}{\partial x^3} = 0 \tag{1}$$

$$D_{11}\frac{\partial^4 w}{\partial x^4} - B_{11}\frac{\partial^3 u}{\partial x^3} = N_x\frac{\partial^2 w}{\partial x^2} \tag{2}$$

where

u	...	inplane displacement
w	...	transverse displacement
A_{ij}	...	extensional stiffness
B_{ij}	...	coupling stiffness
D_{ij}	...	bending stiffness, and
Nx	...	inplane force per unit width

Differentiation of the first equation with respect to x and substitution into the second equation yields:

$$\frac{\partial^4 w}{\partial x^4}+\frac{A_{11}N_x}{B_{11}^2-A_{11}D_{11}}\frac{\partial^2 w}{\partial x^2}=0 \qquad (3)$$

$$N_x=\frac{B_{11}^2-A_{11}D_{11}}{A_{11}}\left(\frac{m\pi}{l}\right)^2 \qquad (4)$$

where

m	...	integer number
l	...	length of the beam.

Graphical results of the calculated buckling force for unit length and buckling force divided by weight for unit length, illustrate that larger buckling forces can be sustained as sandwich beam thickness is increased by any of the constituents. The Foam core supports a greater load than the honeycomb cores (HEXCEL 1 and 2), and the Hexcel 2 core has a load carrying advantage over the Hexcel 1 core. Additionally, results indicate that when weight is a consideration, there is an optimum Ti6Al-4V thickness of approximately 0.10 inches. (Figures 1-6)

As prototype full thickness testing may not be possible due to laboratory machinery limitations, given the calculated buckling loads, a scaled down test specimen is recommended.

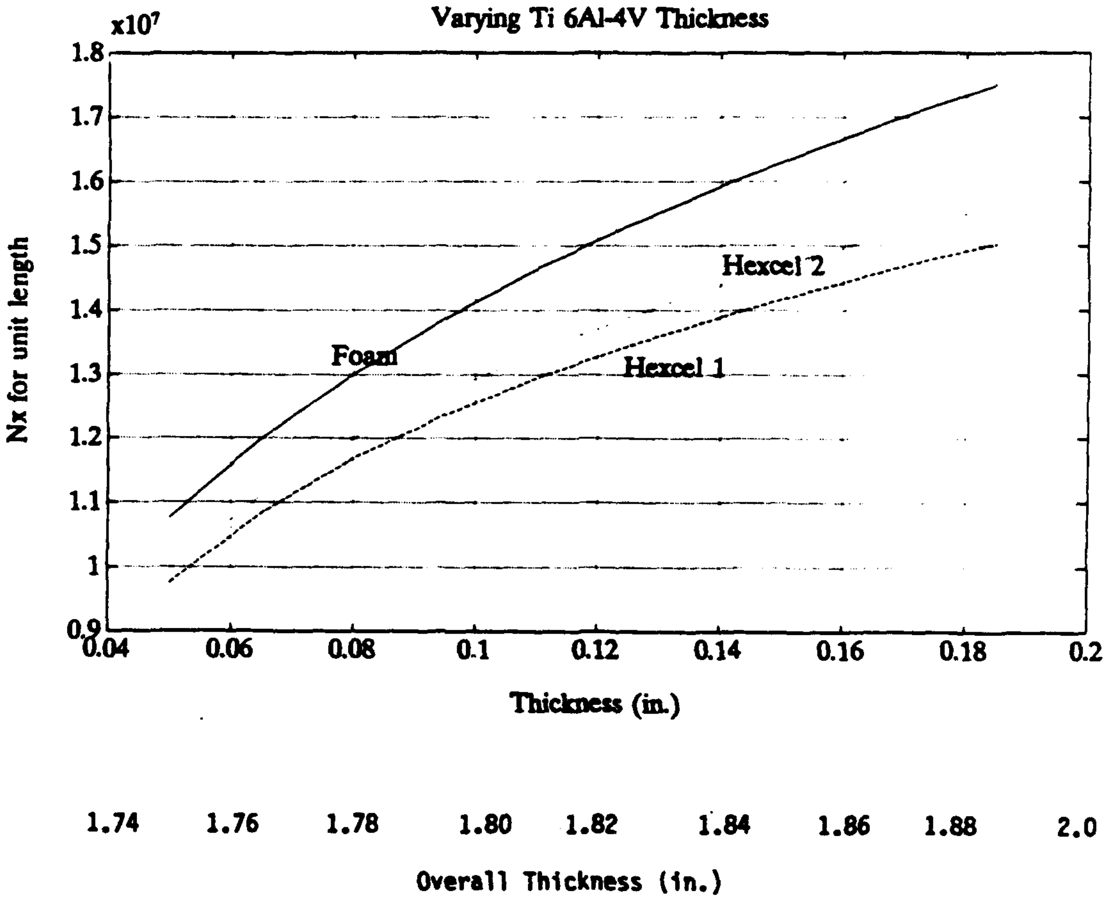

Figure 1. Theoretical Buckling Force With 1.5" Cores and 0.2" GRP Skin Thickness.

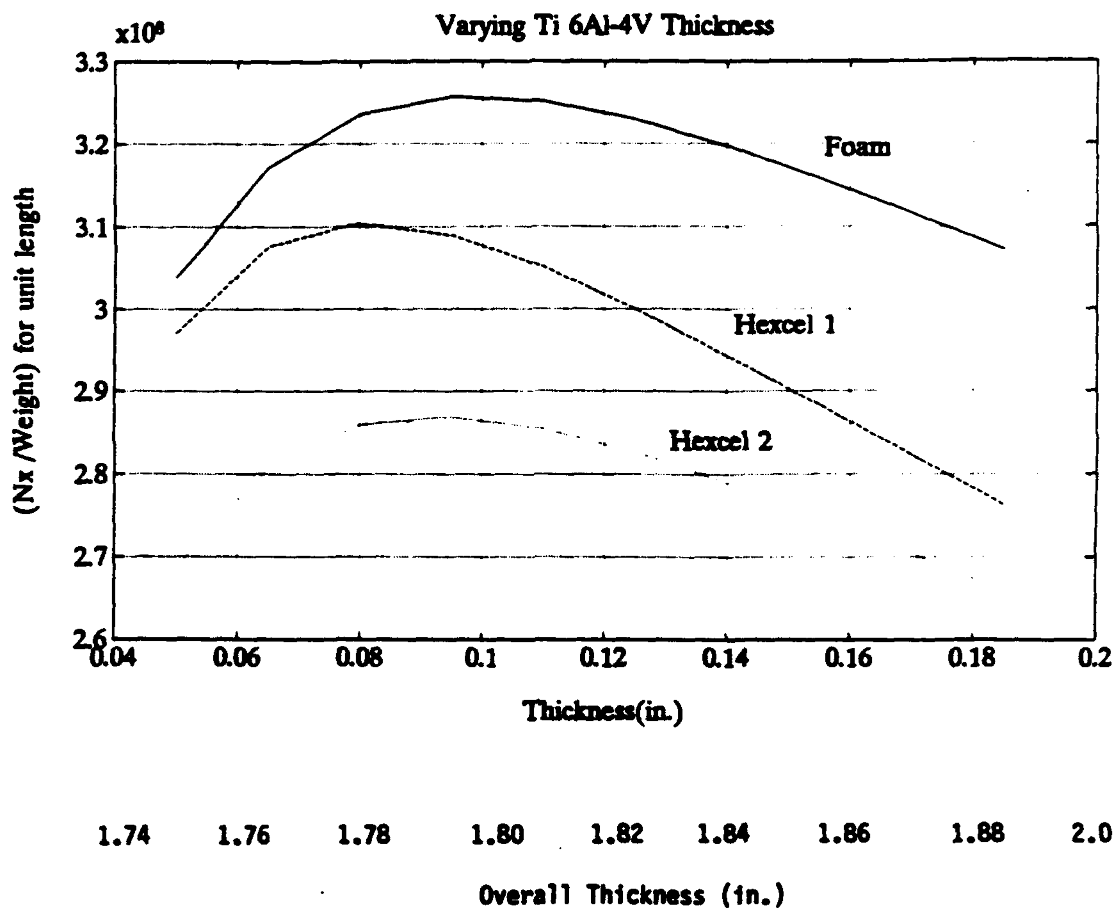

Figure 2. Buckling Force Per Weight With 1.5" Cores and 0.2" GRP Skin Thickness.

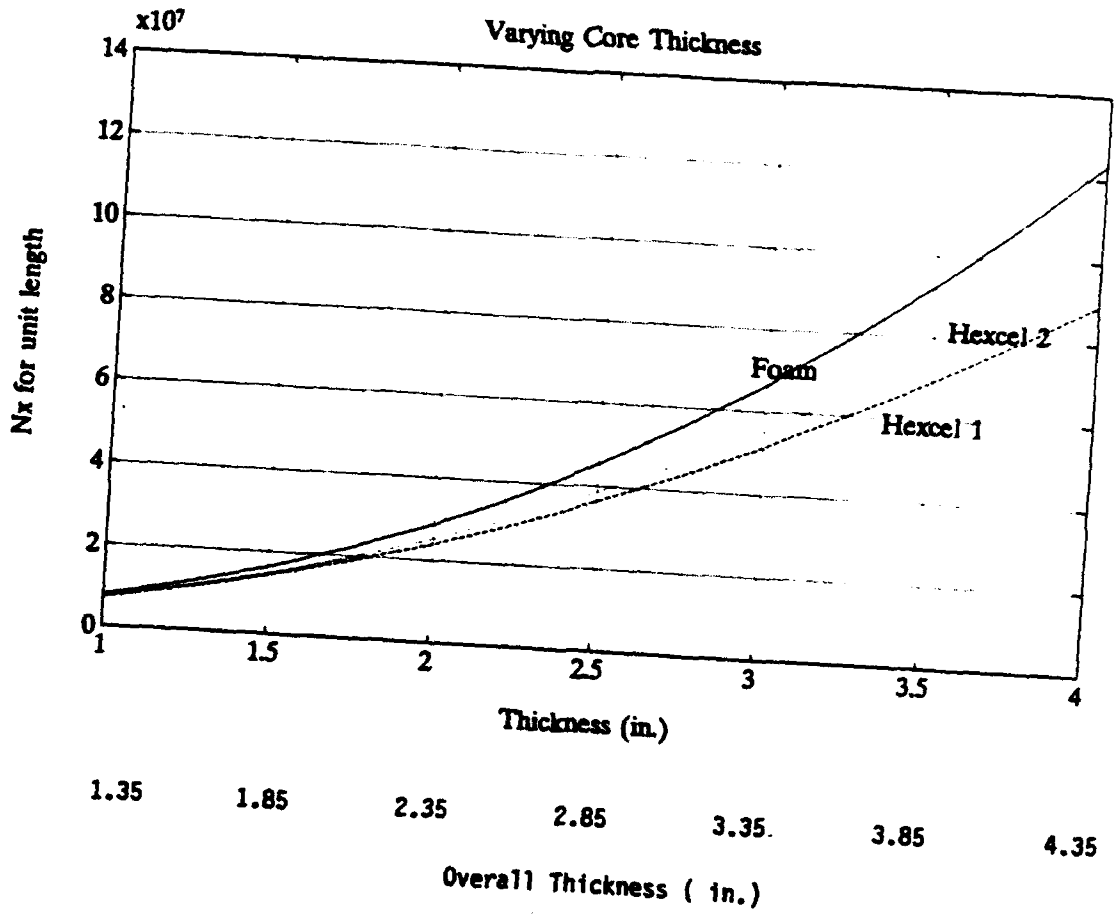

Figure 3. Theoretical Buckling Force With Ti6-4 and GRP Skin Materials Fixed at 0.15" and 0.2", Respectively.

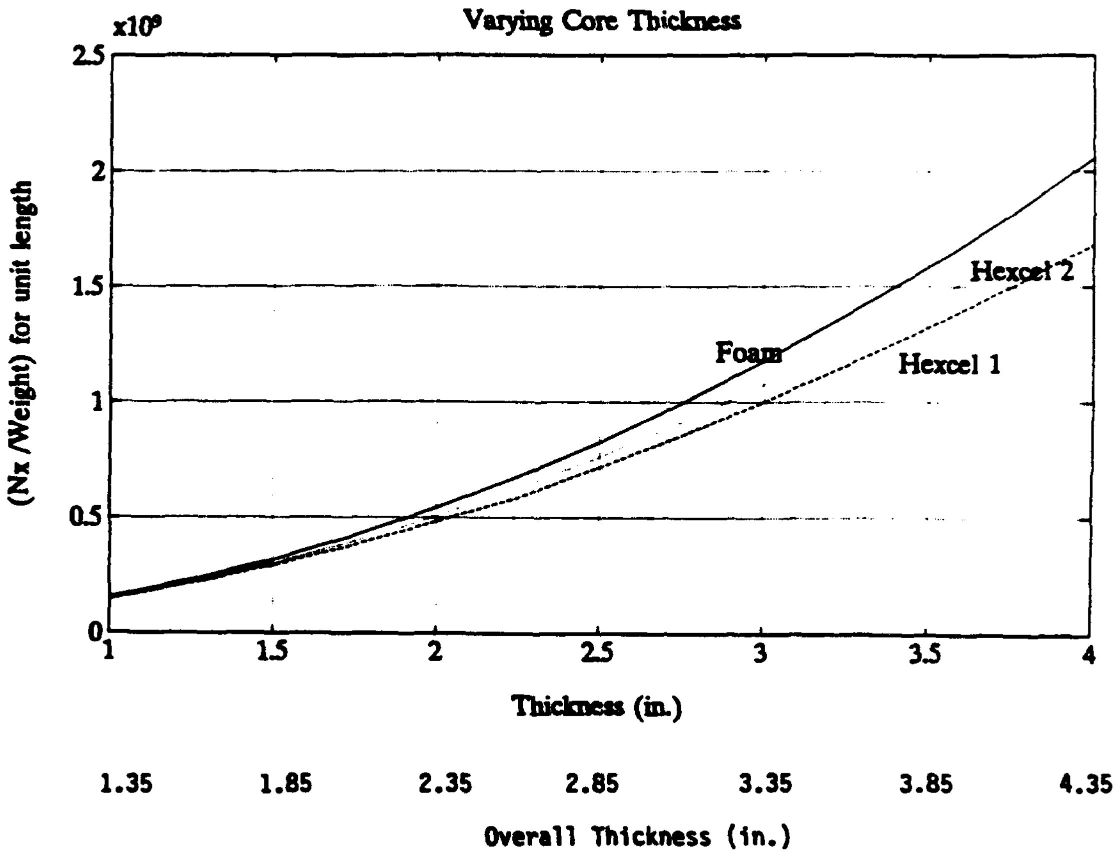

Figure 4. Buckling Force Per Weight With Ti6-4 and GRP Skin Materials Fixed at 0.15" and 0.2", Respectively.

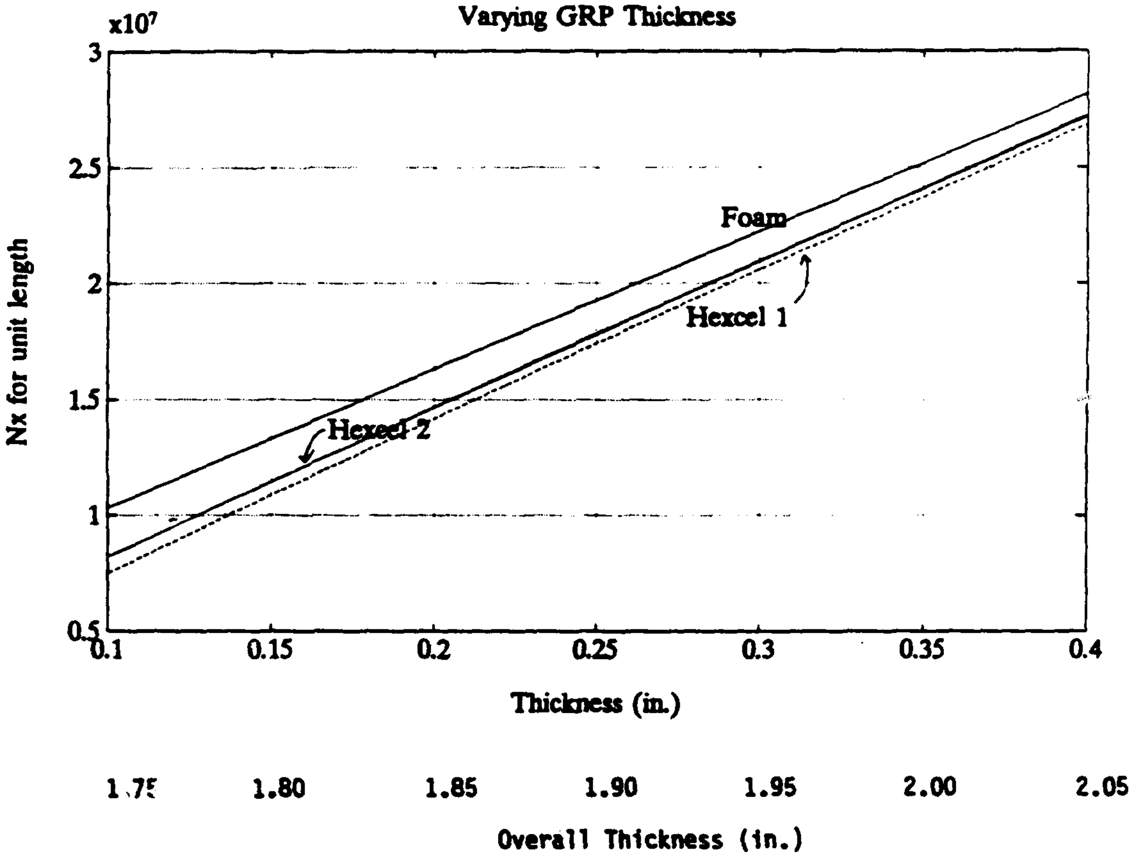

Figure 5. Theoretical Buckling Force When Cores are 1.5" and Ti6-4 0.15".

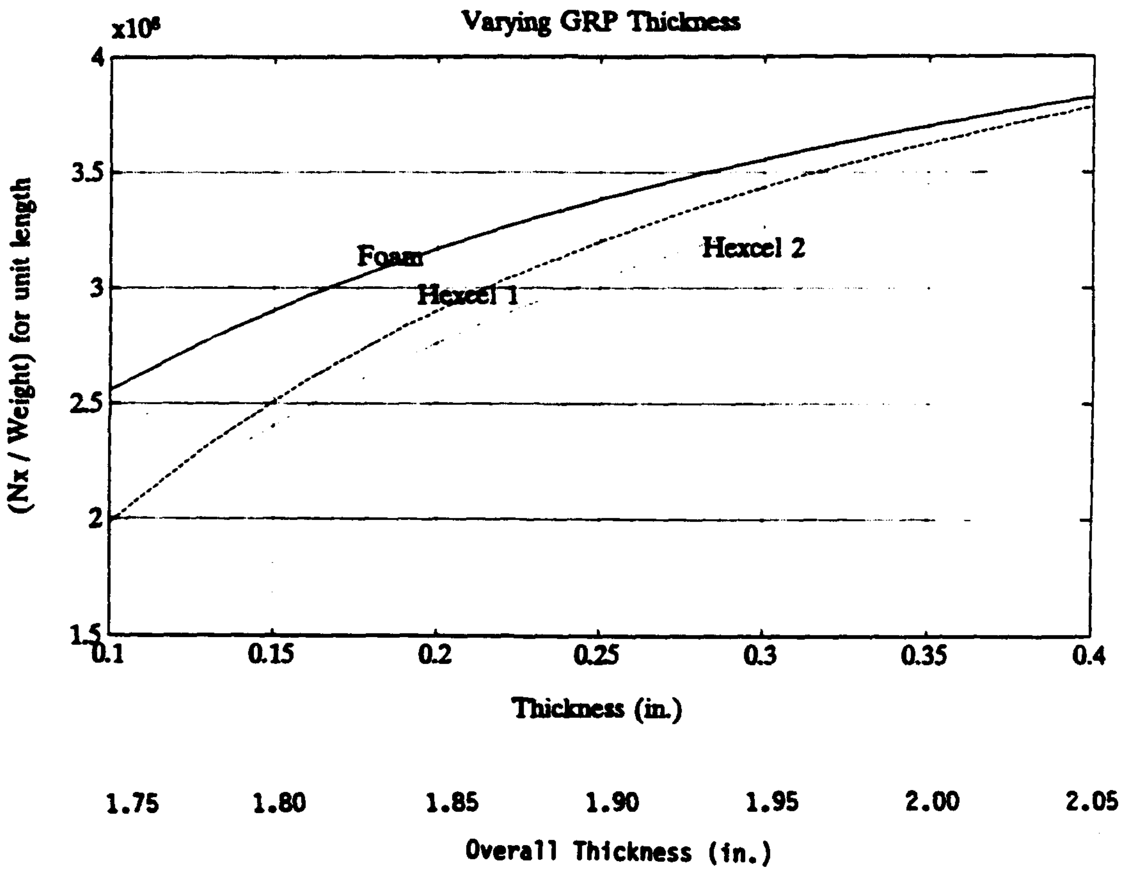

Figure 6. Buckling Force Per Weight When Cores are 1.5" and Ti6-4 is 0.15".

The coupon dimensions should be varied to investigate what is the most significant failure mode among four possible compressive failure modes: general buckling, shear crimping, dimpling of facings, and wrinkling of the faces. (Figure 7) Additionally, the size effect should be studied to predict the failure load in the prototype.

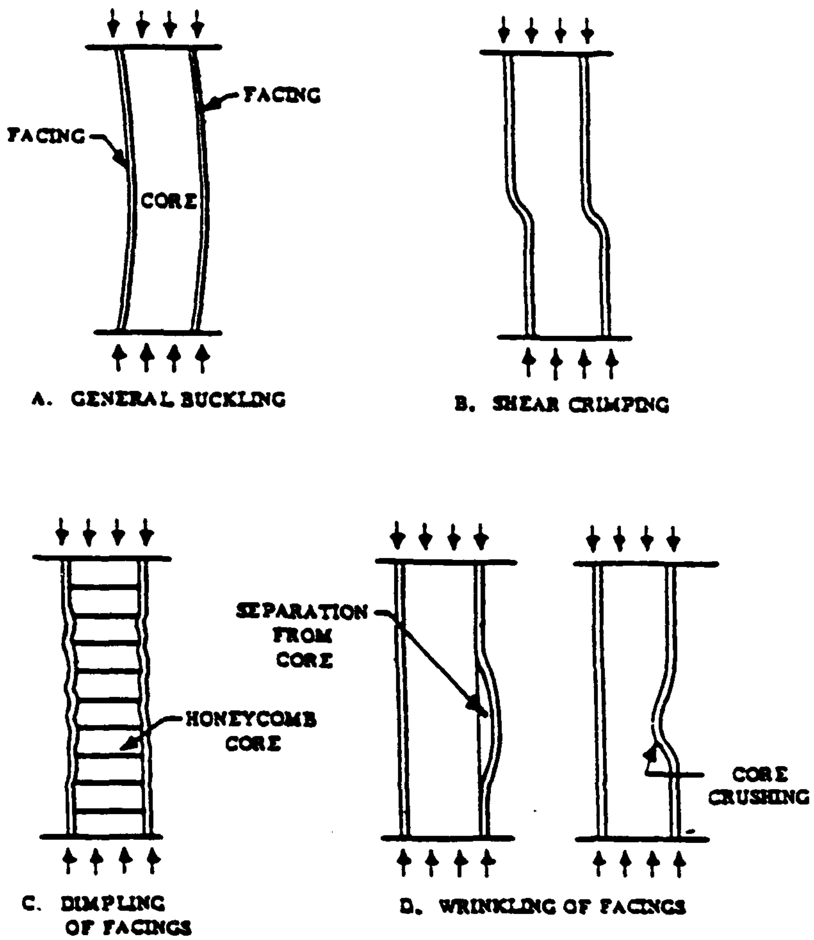

Figure 7. Possible Modes of Failure of a Sandwich Composite Under Edgewise Loads.
Source: MIL STD 401B Sandwich Constructions and Core Materials; General Test Methods 26 Sep 1967.

III. EXPERIMENTAL PROCEDURES

This section provides a detailed description and illustrations of the experimental apparatus used, laboratory conditions, and testing procedures.

A. APPARATUS

All tests were conducted in the Mechanical Engineering building at the Naval Postgraduate School, Monterey, California, in an ambient temperature of 22.5 ± 2.0°C with an average relative humidity equal to 41 ± 5%.

Axial compressive load was applied using the Riehle Material testing machine, capacity 120,000 pounds (10-50 pounds per division). A testing fixture was designed to provide simply supported end conditions on the loaded surfaces of each beam; the unloaded side surfaces were unconstrained. The simply supported condition was accomplished using two 3 inch diameter, 11 inch long, Rycase (1117) low carbon, high manganese steel round shafts machined with keyways for holding specimens and shims. Each shaft was mounted in two Dodge unisphere 3" pillow blocks, part #041482. The shafts were free to rotate 360° in the bearings. The bearings were bolted to aluminum plates fixed to the Riehle testing unit. Figures 8-11 illustrate the fixture components.

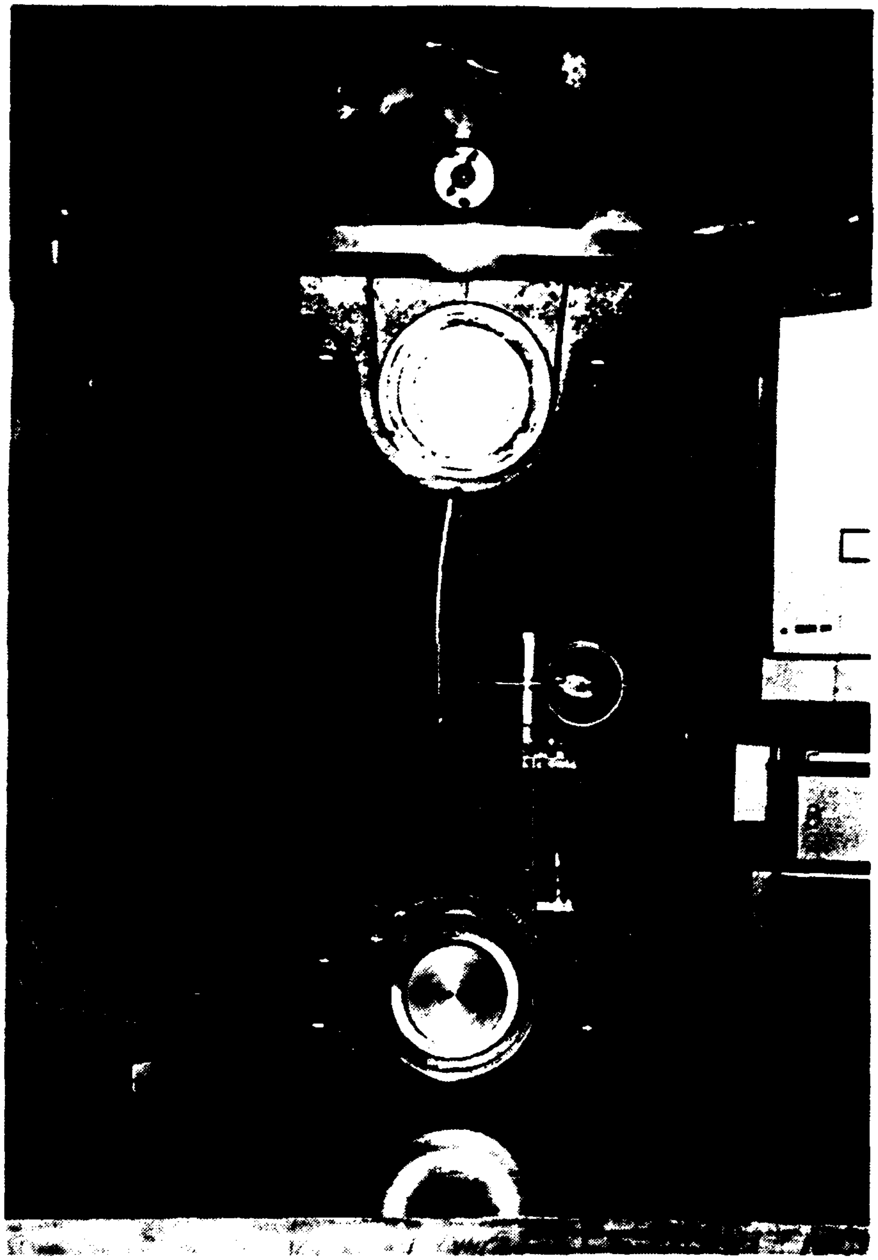

Figure 8. Unbalanced, Sandwich Composite Beam Mounted in Testing Fixture.

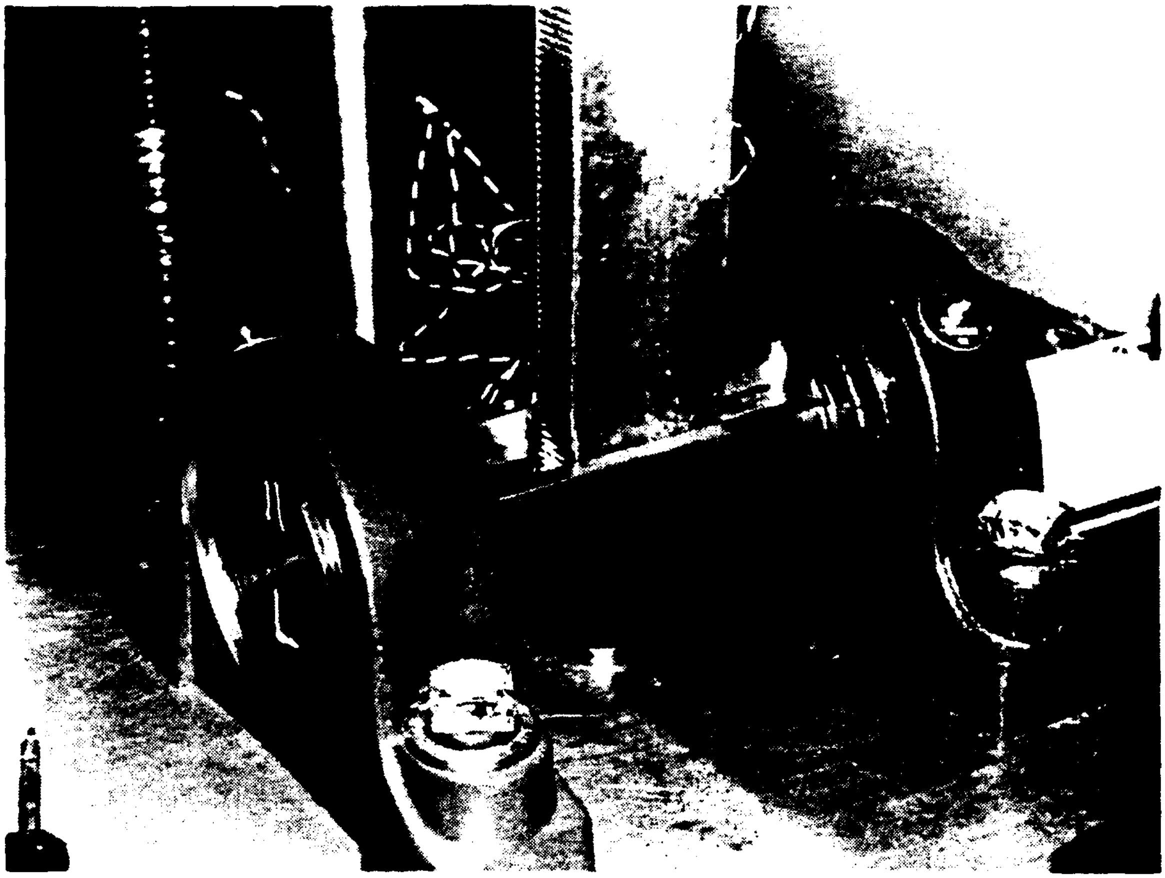

Figure 9. Unbalanced, Sandwich Composite Mounted in a Freely Rotating Shaft Achieves Simply Supported Boundary Condition.

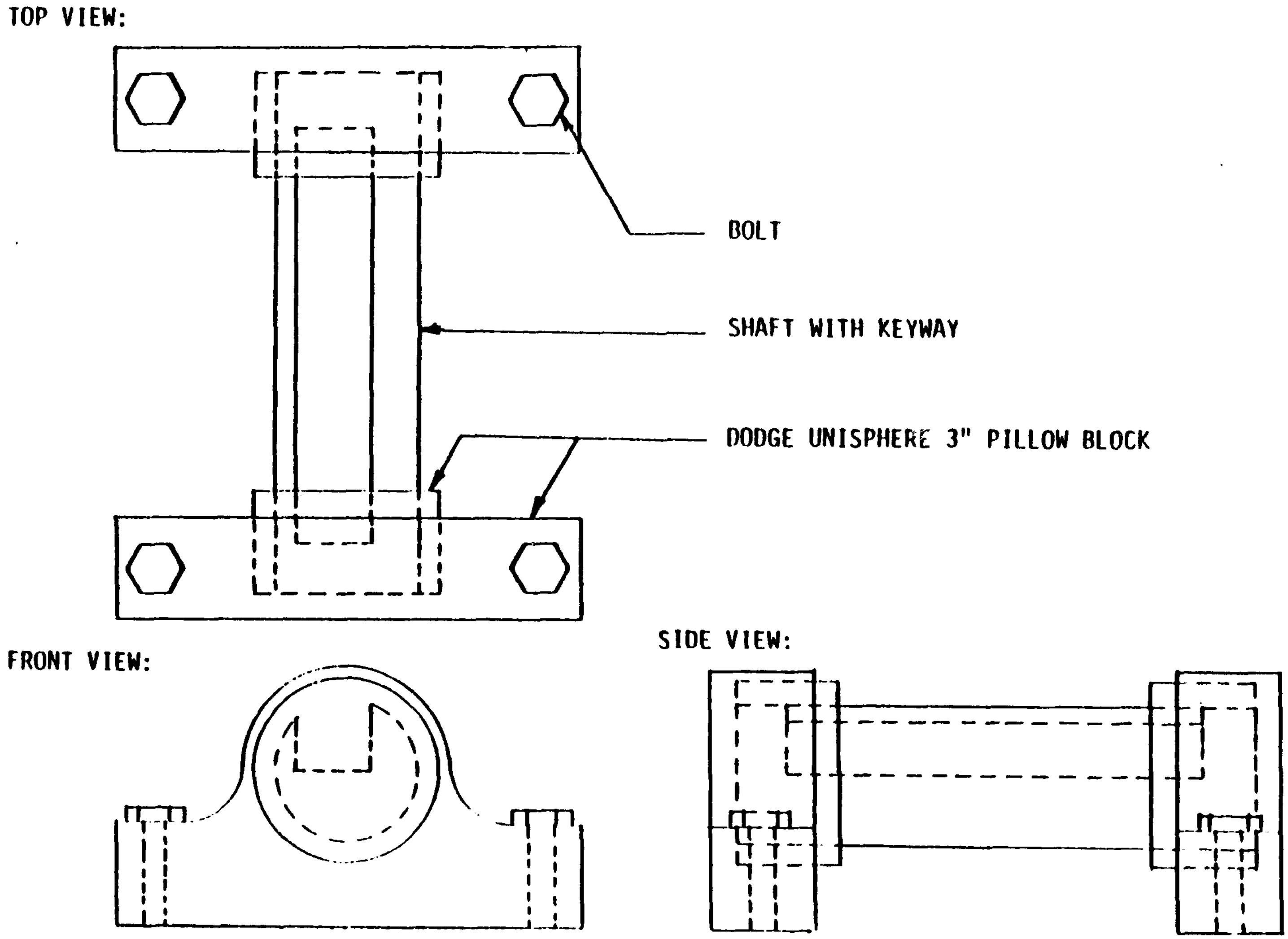

Figure 10. Details of Shaft Design Showing (a) Top, (b) Front, and (c) Side Views.

DETAILED FRONT VIEW:

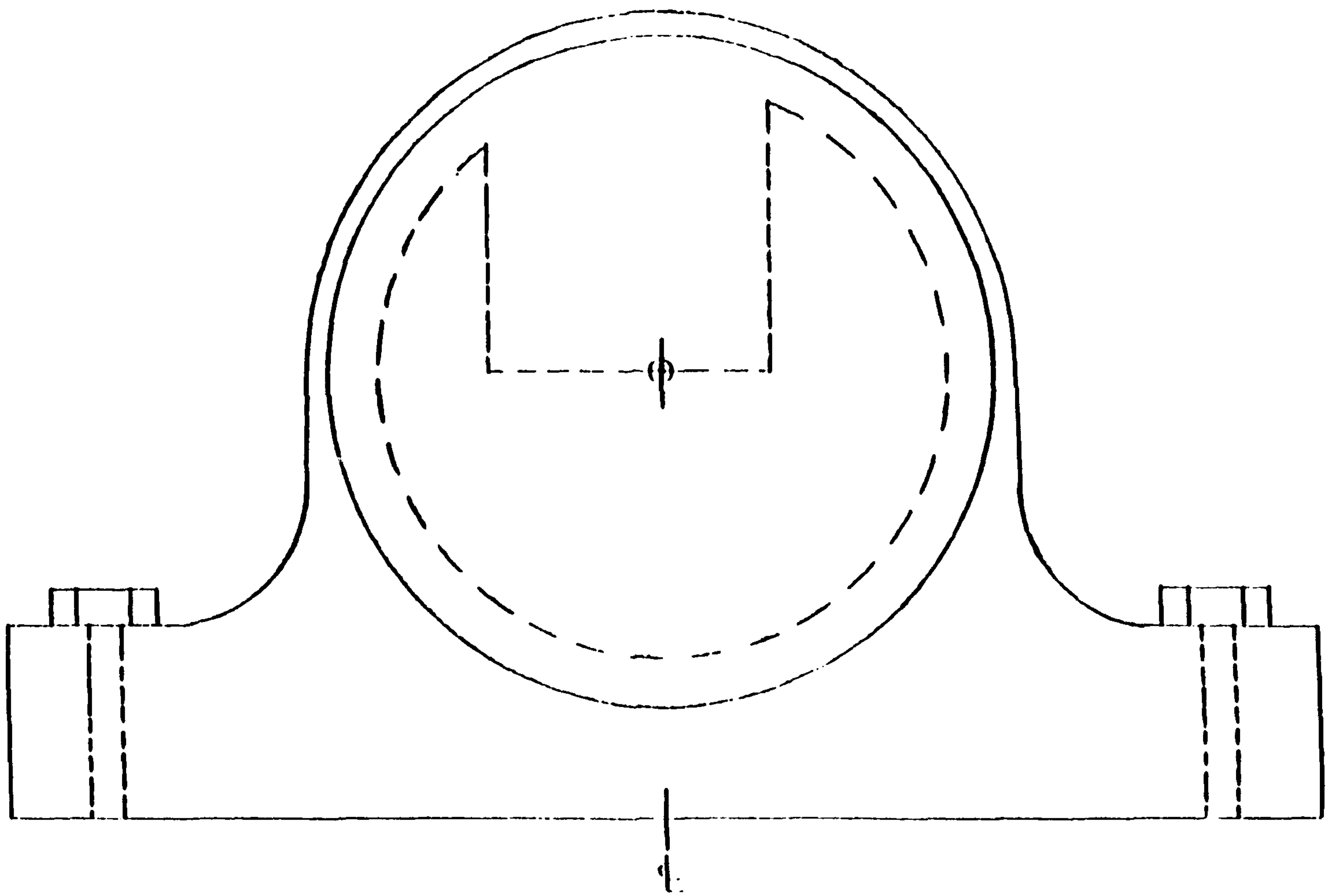

Figure 11. Detailed View of Simply Supported End Condition.

The specimens were nominally three inches wide and varied in length and thickness. Skin materials were tested separately, prior to the unbalanced sandwich constructions. The Titanium 6 aluminum -4 vanadium (Ti6-4) skins were instrumented with 1/4 inch CEA-06-250UN-350 precision strain gages, gage factor 2.100 ± 0.5%. Two strain gages were mounted side by side at the center of the length of the specimen. Additional gages were mounted at 1/4 and 3/4 length distances on the 36 inch specimen. The composite skins were instrumented similarly with 1/4 inch CEA-13-250UN-350 precision strain gages, gage factor 2.120 ± 0.5%. The unbalanced sandwich composite samples were outfitted with gages located at several positions on both skins, as noted on data sheets. Gage outputs were connected to a Measurements Group SB-10 Switch & Balance Unit, and readouts provided by Measurements Group P-3500 Strain indicator, in microstrain.

Specimens were held in place in the shaft keyways using aluminum blocks and shims of various sizes. A distance transducer, model #DV301-6020-111-1110 was mounted vertically and attached to the upper aluminum base plate to measure axial contraction in inches. Center deflection was measured for the unbalanced, sandwich construction samples using a Starrett 1.000" dial indicator. Temperature and relative humidity were determined utilizing the Vaisala HMI 33 probe and transducer with digital readout.

B. PROCEDURE

Specimens and shims were centered longitudinally in the shafts using three inch spacer blocks. Specimens were shimmed to fit snugly into the fixture, but without causing grabbing of the specimen ends while loading.

Testing of both the skin materials and composite sandwich construction was manually load controlled. Low speed loading was incrementally halted to allow recording of strain, displacement and deflection. Specimens were loaded beyond the point of instability to ensure proper data collection.

C. MATERIAL PROPERTIES

In order to verify the Young's moduli of the skin materials, tensile testing was conducted using the MTS 810 Material Test System.

Three titanium alloy coupons were machined and tested in accordance with ASTM E8-91. The average value of Young's modulus was 15.4×10^6 psi.

ASTM D3039-76 (reapproved 1989) was used as a guideline and followed as closely as possible to test the GRP skin material. The woven glass material does not strictly meet the scope of this ASTM, chosen as the most appropriate. End tabs were manufactured using four tapered layers of the composite itself. One test was accomplished, yielding a value for Young's modulus of 3.0×10^6 psi.

D. EXPERIMENTAL ERROR

The largest experimental error occurred when measuring small compressive loads (less than 200 pounds). Since the Riehle machine capacity is 120,000 pounds, it is less sensitive in this range. Individual experimental errors are discussed in Chapter IV.

IV. SKIN MATERIAL ANALYSIS

A. Ti6Al-4V

The buckling loads for 13 titanium alloy skin samples of various lengths are listed in Table 2. The nominal thickness of each sample was .106 inches and the nominal width 3.00 inches. Materials, as received, exhibited some initial curvature. The overall buckling load was determined by measuring the maximum load in pounds force that the sample would support. Plots of the Load versus Strain or Load versus Displacement clearly depict this. (see Figures 12 and 13). The predicted buckling load for an isotropic column that is simply supported is given by Euler:

$$P_{crit.} \frac{\pi^2 EI}{L^2} \qquad (5)$$

where

$P_{crit.}$ = critical buckling load in pounds force
E = Young's modulus (15.5×10^6 p.s.i.)
I = moment of inertia = 1/12 x width x thickness^3
L = length in inches

Results illustrated in Figure 14 show good agreement between experiment and theory and lend credibility to the test fixture to achieve simply supported boundary conditions on the loaded edges. Experimental errors were as low as three percent for the shortest samples and as great as 20 percent for the longest length.

This is a result of the Riehle testing machine's decrease in accuracy below about 200 pounds force.

TABLE 2. TITANIUM 6Al-4V SKIN MATERIAL BUCKLING LOADS

Buckling Load (pounds Force per unit width)

Length (inches)	Experimental Limit Load	Average	Theoretical Buckling Loads	Percent Error
8	252,260	256	244	4.9%
10	163	163	157	4.0%
12	119,125	122	108.8	12.0%
14	87	87	80	8.8%
16	58, 60	59	61	3.6%
18	41.7	41.7	48.4	13.8%
20	32.3, 50	41.2	39.2	5.1%
36	26.7, 6.7	16.7	12.1	38.0%

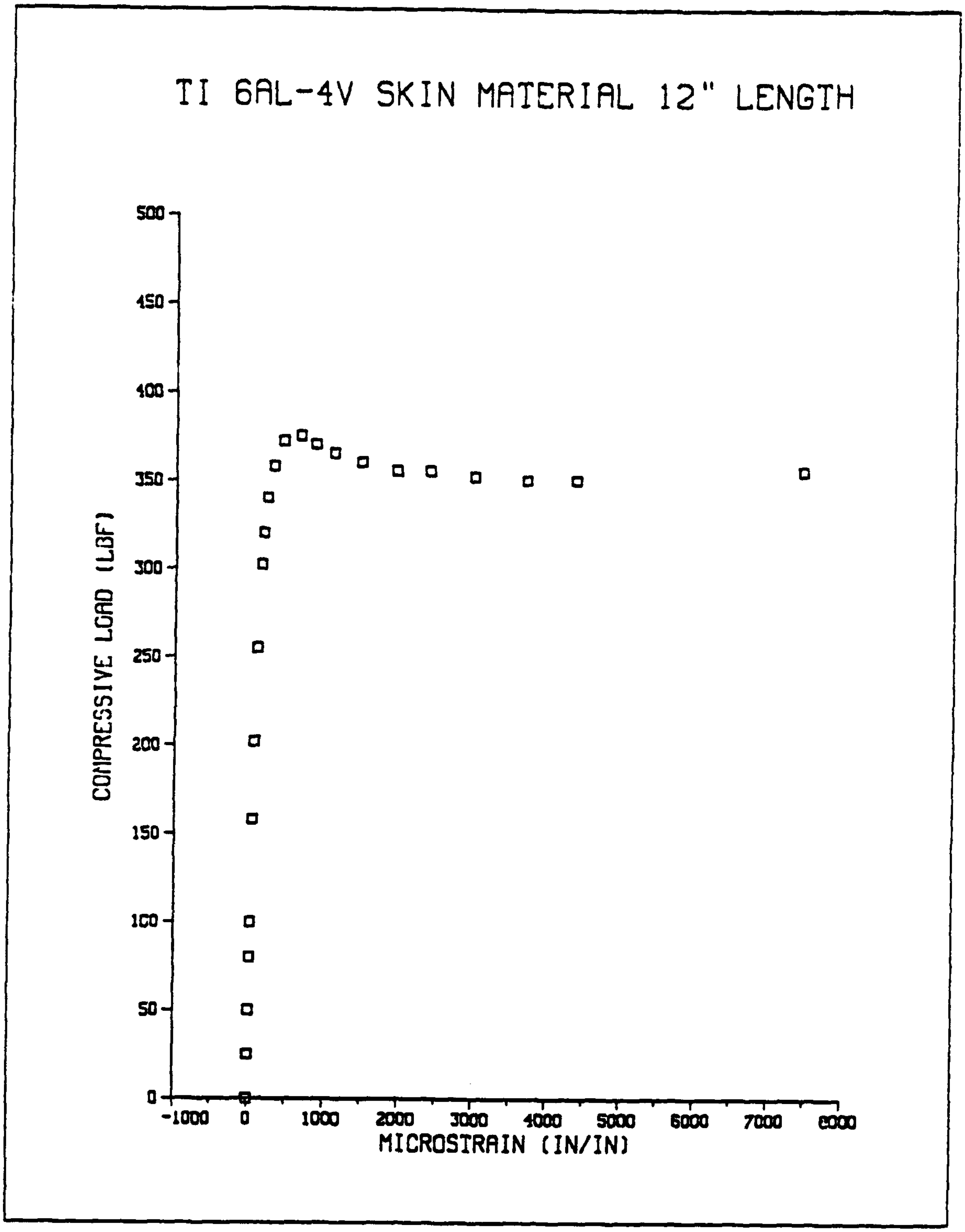

Figure 12. Load Versus Strain for Ti6-4 Skin Material Buckling Instability Test.

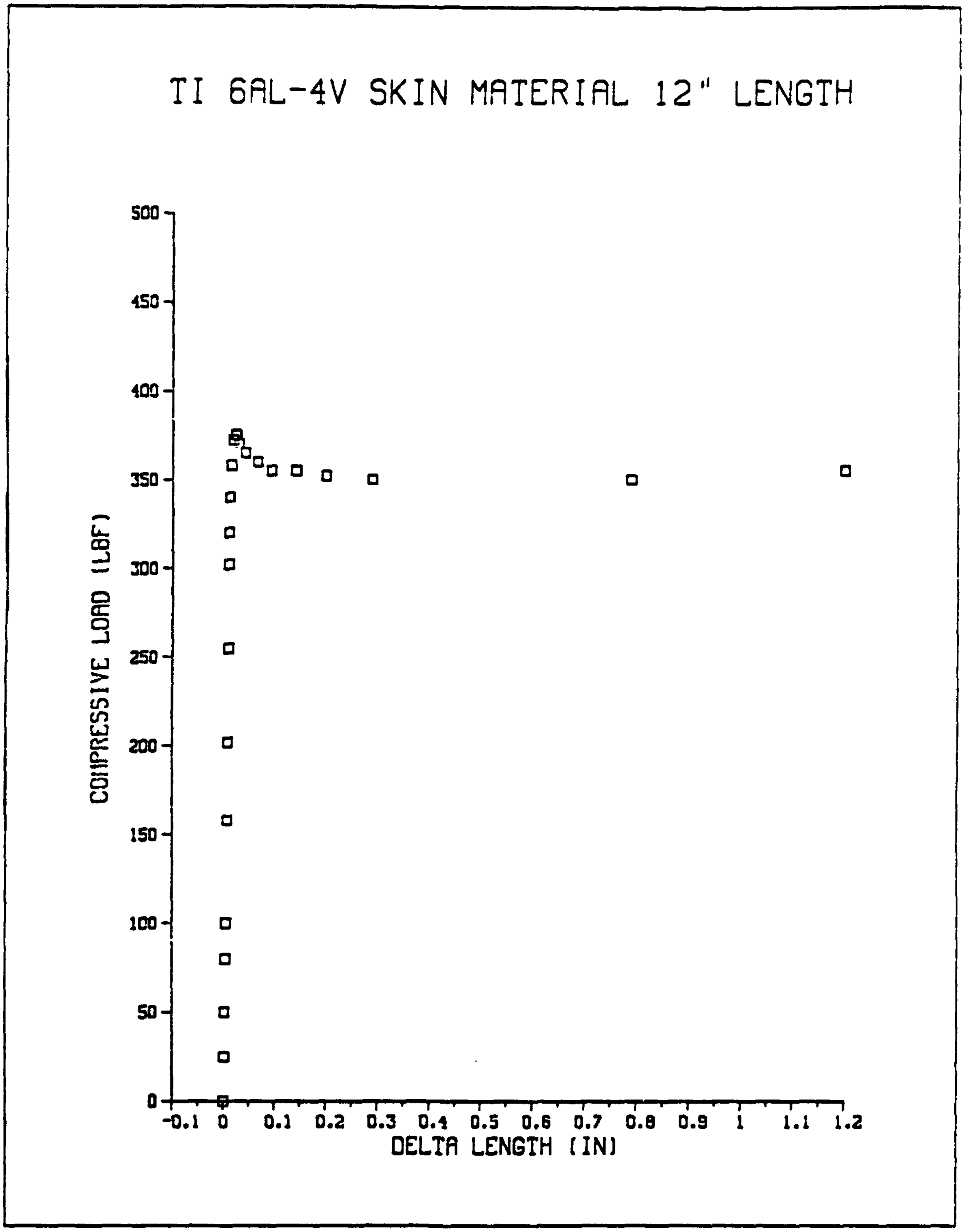

Figure 13. Load Versus Axial Displacement for Ti6-4 Skin Material Buckling Instability Test.

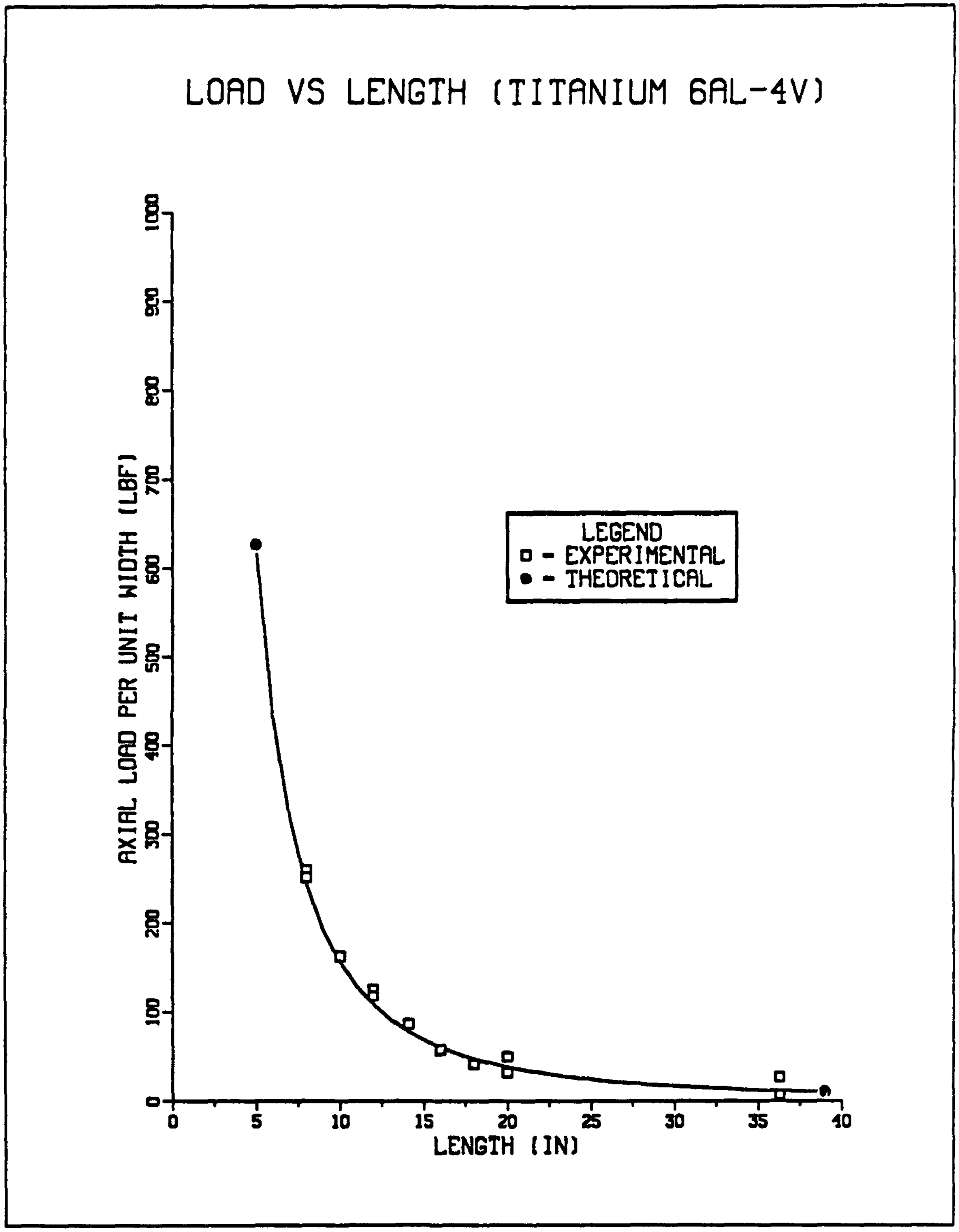

Figure 14. Buckling Instability Test Results for Ti6-4 Skin Material.

B. COMPOSITE

$0^0/90^0$ woven glass in vinyl ester resin laminae were used to construct the outside skin of the unbalanced sandwich construction. Two laminae oriented at $\pm 45^0$ were placed between two $0^0/90^0$ direction laminae, to form a quasi-orthotropic composite laminate.

Sixteen samples were tested and results listed in Table 3. The nominal thickness of the composite skin samples was .081 inches and the nominal width of 3.00 inches. All samples were initially twisted slightly from a flat plane. The predicted buckling load was calculated using Vinson's elastic governing equations for orthotropic laminates and beam theory [Ref. 3].

$$P_{crit} = \frac{\pi^2 D_{11}}{L^2} \tag{6}$$

where

P_{crit} = critical buckling load (pounds force)
D_{11} = bending stiffness in the direction of the length (pounds force-inch)
L = length (inches)

Theoretical and experimental results are illustrated in Figure 15. Typical Load versus Strain and Load versus Displacement curves are illustrated in Figures 16-17.

TABLE 3. WOVEN GLASS IN VINYL ESTER SKIN MATERIAL BUCKLING LOADS.

Buckling Load (pounds Force per unit width)

Length (inches)	Experimental Limit Loads	Average	Theoretical Buckling Loads	Percent Error
7	33,30	32	29.1	10[1]
8	27,19,30,41	29	22.3	23[1]
9	28,16	22	17.6	30[1]
10	12,25,20	19	14.3	32.8[1]
11	9	9	11.8	23.7
12	13,8	11	10	10
12.8	7	7	8.7	19.5
19.9	4	4	3.6	11

Note 1. If theoretical values are multiplied by $1/(1-\upsilon^2)$ to account for shortness of beam, these errors reduce to 2.1, 15.7, 11.2 and 18.2 percent respectively. Here υ is Poisson's ratio.

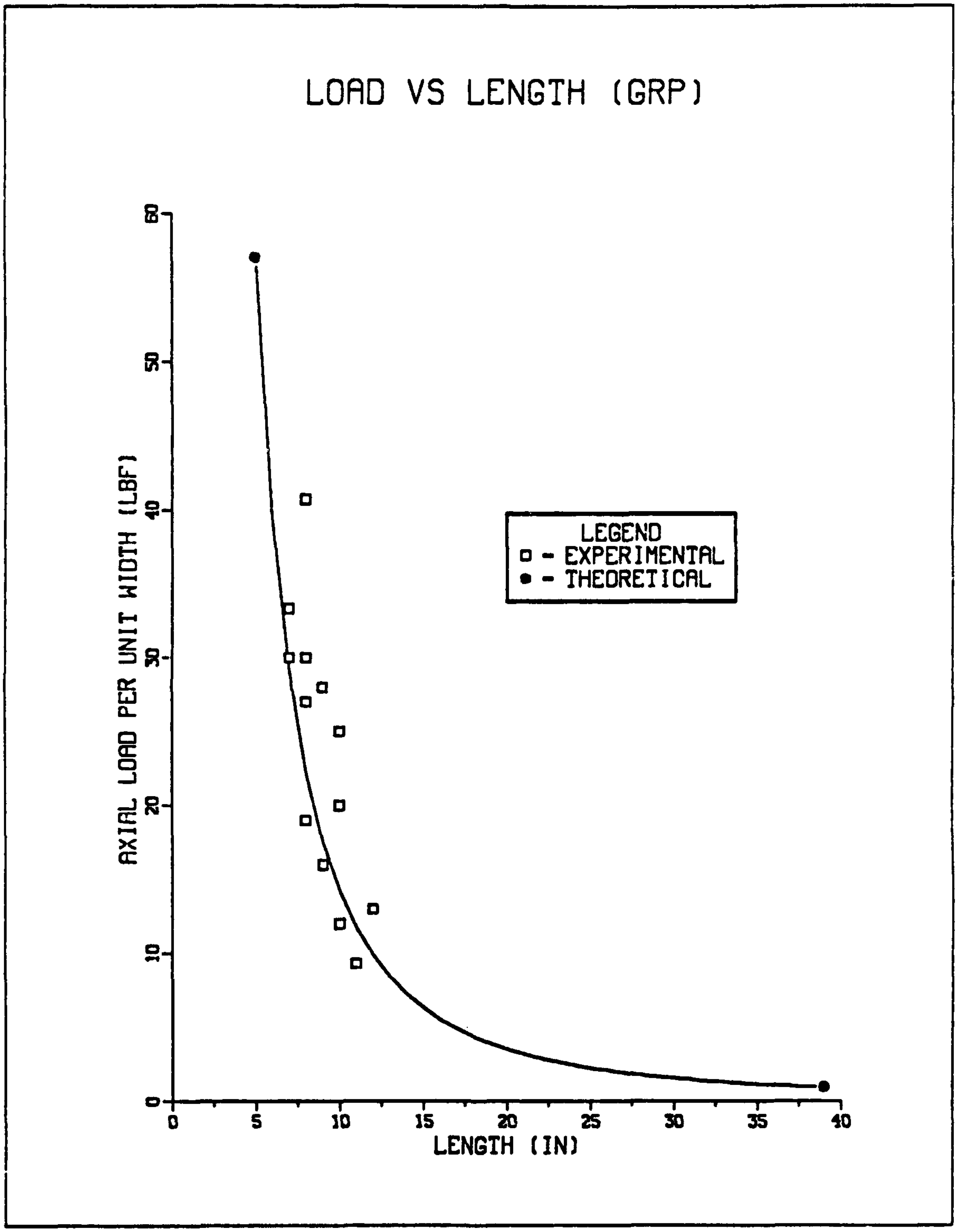

Figure 15. Buckling Instability Test Results for GRP Skin Material.

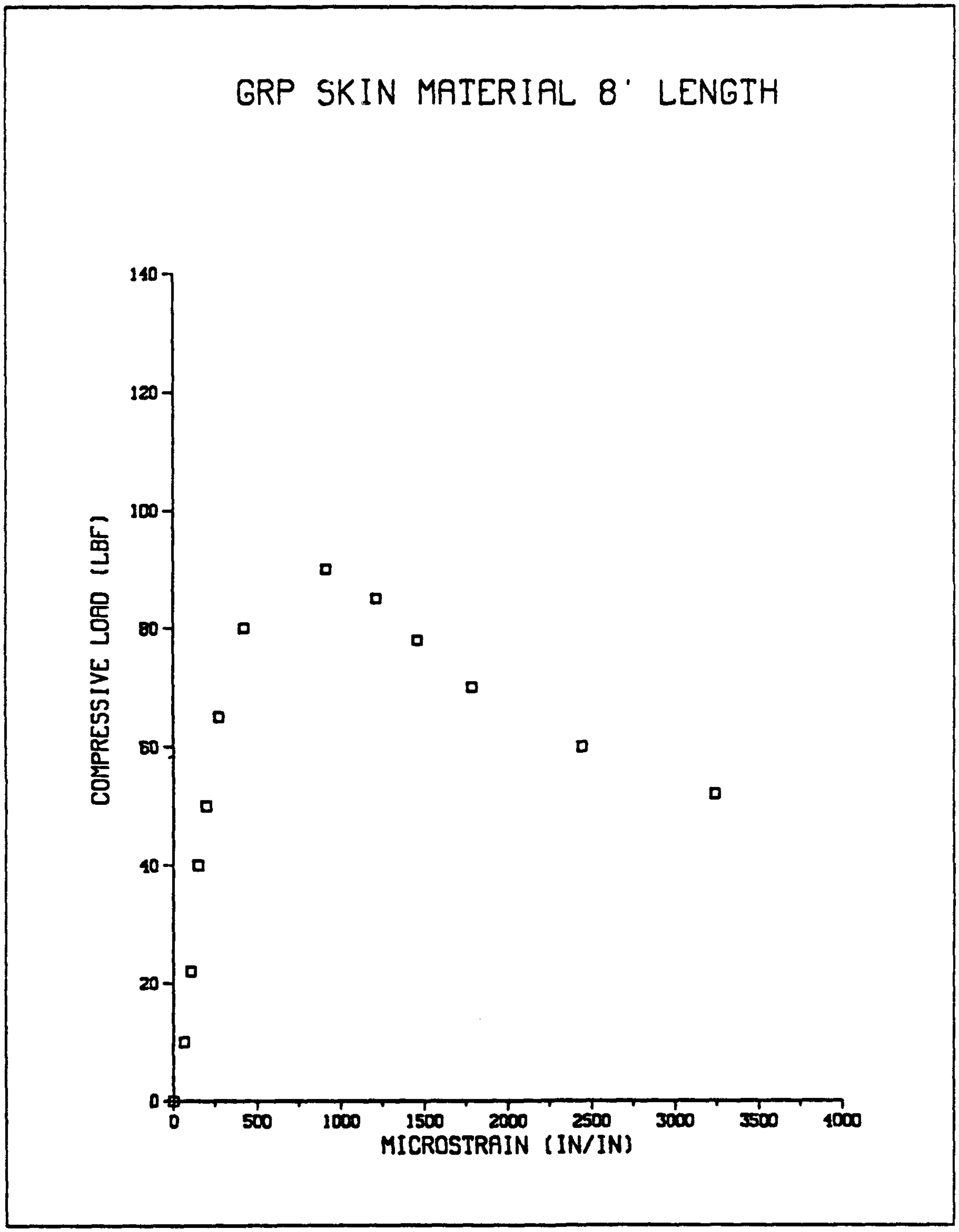

Figure 16. Load Versus Strain for GRP Skin Material Buckling Instability Test.

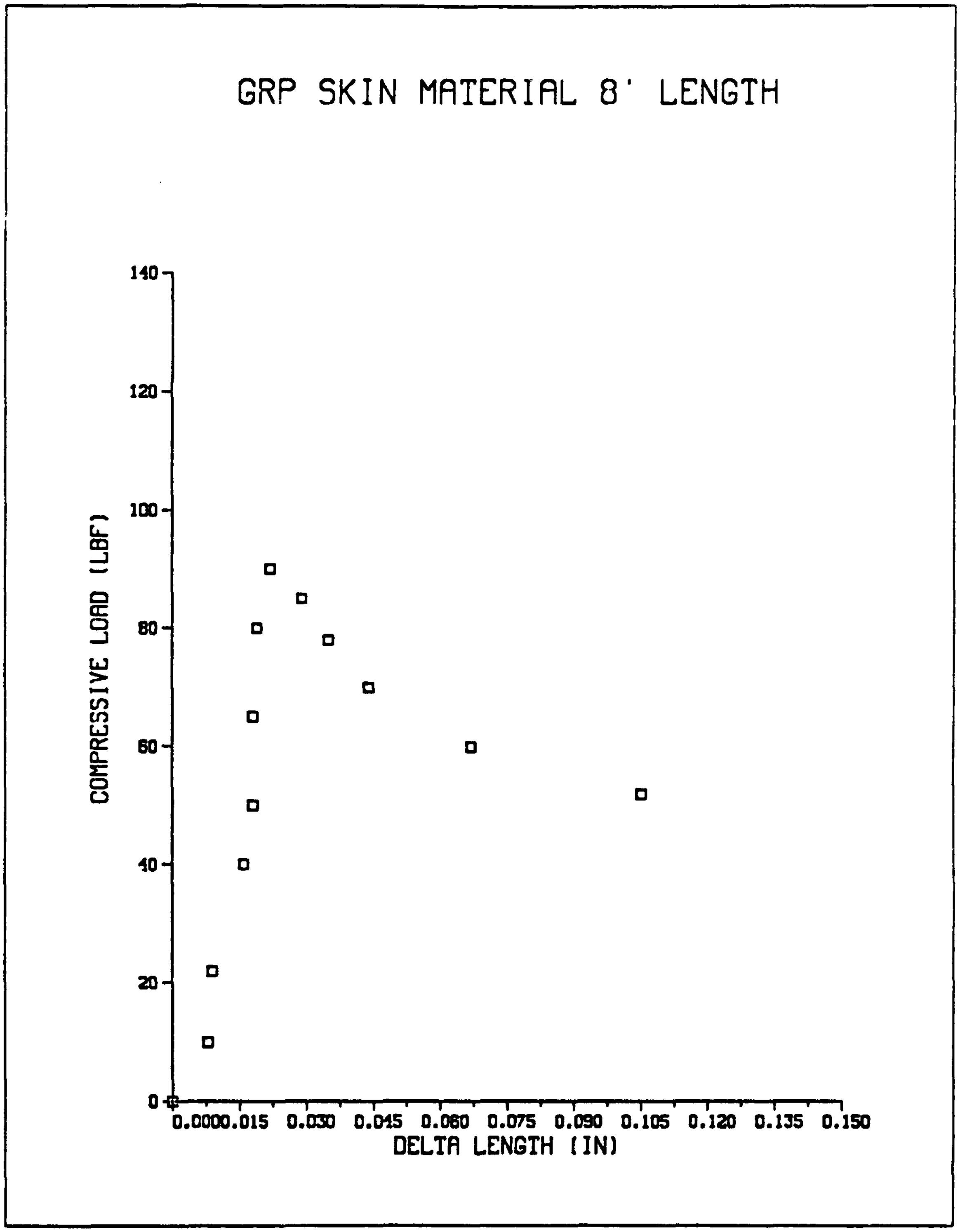

Figure 17. Load Versus Axial Displacement for GRP Skin Material Buckling Instability Test.

The stiffness of the composite skin material is about one-tenth of that of the titanium, and as noted previously, the experimental uncertainty increases with decreasing length. Poisson's effect may influence the buckling load of samples ten inches or less in length where specimen geometry becomes more like a plate than a beam. A factor of $1/(1-\nu^2)$ is used, therefore, in determining the theoretical buckling load for these shorter lengths.

In summary, analysis of the limit loads of the titanium alloy and GRP composite skin materials showed that theoretical predictions for a simply supported beam configuration were valid and promoted confidence and reliability in the testing fixture as designed.

V. UNBALANCED, SANDWICH COMPOSITE ANALYSIS

A. SPECIMEN DESCRIPTION

The unbalanced, sandwich composite samples tested were manufactured at the Naval Surface Warfare Center, Carderock Division. The skin materials were the same as those analyzed previously. The core was an aramid fiber/phenolic resin 1/8" hexagonal cell honeycomb[1]. Material properties are listed in Table 4. The overall thickness of specimens was nominally 1.2 inches, the width nominally 2.75 inches, and length of 36 inches. It is mentioned here that an initial curvature, approximately .015 inches, appeared in of all samples received. Although Suarez, et al, cite initial waviness as a mechanism to induce core shear failure [Ref. 5], this does not appear as the primary failure mode. The limit loads achieved are much higher than Suarez' predictions, even when the titanium material properties are used in the calculation because the equation was developed for a balanced (symmetric) sandwich composites.

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[1]HRH-10, manufactured by HEXCEL.

TABLE 4. MATERIAL PROPERTIES OF UNBALANCED, SANDWICH COMPOSITE CONSTITUENTS.

	Ti6Al-4V[1]	GRP[2]	HRH-10 Core[3]
E (psi)	15.5×10^6	3.0×10^6	28×10^3
G_{12} (psi)	6.1×10^6	1.2×10^6	8.6×10^3
G_{13} (psi)	-	-	4.7×10^3
ν	.342	.33	-
t (inches)	.106[4]	.081[4]	1.0

[1]Values obtained from Metals Handbook, 10th edition, Volume 2

[2]Average values obtained experimentally by the Naval Surface Warfare Center. Young's modulus, E, decreases with increased strain

[3]Values obtained from Hexcel Data Sheet 4000

[4]Based on previous average values of each skin material

Four original lengths of 36 inches were tested; the rest of the sandwich beams were cut into shorter lengths of 25, 18, and 12 inches. A total of eight tests were performed.

Back to back strain gages (one on each face) were mounted at the sample centers, one-quarter length from the end, one-eight length from the end, and various paired combinations as illustrated in Figures 18. The axial displacement and center transverse deflection were also monitored.

The samples were mounted in the test fixture so that loading could be applied on or off the neutral axis. The neutral axis was calculated, neglecting the effect of gluing materials, to be approximately .085 inches inward from the Ti6-4 skin and core interface. An axial compressive load was manually applied, as described in the skin material testing.

B. LIMIT LOAD PREDICTIONS

Initial predictions of the general buckling failure loads for the overall sandwich structure were obtained utilizing the equations presented in Chapter II, equation (4) and the material properties listed in Table 4. Glue thicknesses were assumed negligible, therefore, these calculated loads were considered conservative. Calculations using the midplane axis differed by only a few percent from those employing the neutral axis as a base line for formulation of the extensional, coupling and bending stiffnesses (A_{11}, B_{11}, D_{11}).

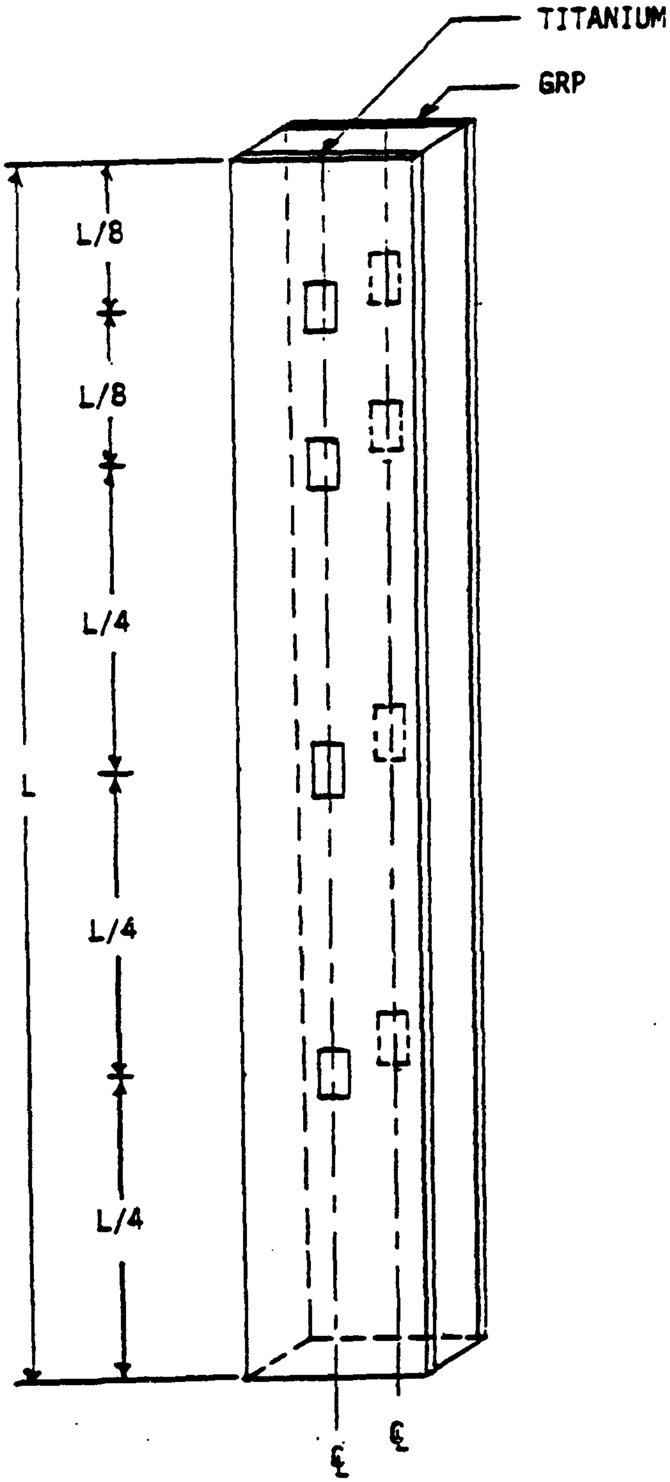

Figure 18. Typical Placement of Back to Back Strain Gages on Sandwich Composite Samples.

The results for four lengths where the edgewise compressive load is applied to the neutral axis are shown in Table 5.

TABLE 5. UNBALANCED, SANDWICH COMPOSITE LOAD APPLIED AT NEUTRAL AXIS

Length (inches)	Theoretical Load (lbf per unit width)	Experimental Load (lbf per unit width)	Ratio Theor to Expt Load
36	1,670	1,330	1.26
24	3,700	2,250	1.64
18	6,590	2,900	2.27
12	14,828	3,650	4.06

In all cases, the observed buckling instability occurred at lower loads than analytically predicted. This difference between the theoretical and experimental loads increased significantly as the sample lengths decreased. Additionally, a second mode of failure, shear crimping, caused the compressive load to drop dramatically at it's onset. Core shearing was a post buckling phenomena in the 36, 24, and 18 inch samples. The 12 inch specimen appeared to buckle and shear simultaneously.

The inclusion of the shear deformation energy into the predictive failure load was considered. The work of Timoshenko and Gere on the effect of shearing force on the critical buckling load for a homogeneous bar [Ref. 4] was

modified to reflect the effect of the unbalanced, skin materials on the core shearing strength.

The strain energy of a beam is given by

$$\Delta U=\frac{1}{(2D)}\int_0^l M^2 dx+\frac{P^2}{2h_c G_c}\int_0^l (v'(x))^2 dx \qquad (7)$$

where

ΔU: Strain energy of the system

M: Bending moment

v: Horizontal beam displacement

D: Flexural Stiffness of Skins ($\frac{1}{3}\Sigma E_i (y^3_{i+1}-y^3_i)$)

h_c: Core Thickness

G_c: Core Shear Modulus along the beam direction

P: Critical buckling load

the external work on the system is:

$$\Delta W=\frac{P}{2}\int_0^l (v'(x))^2 dx \qquad (8)$$

Assume the first buckling mode -

$$v(x)=a\sin\left(\frac{\pi x}{L}\right) \qquad (9)$$

the energy method gives -

$$\frac{P^2L}{4D}a^2 + \frac{\pi^2P^2}{4h_cG_cL}a^2 = \frac{\pi^2P}{4L}a^2 \tag{10}$$

The Buckling Load including core shear deformation is

$$P = \frac{\dfrac{\pi^2D}{L^2}}{1 + \dfrac{\pi^2D}{h_cG_cL^2}} \tag{11}$$

Table 6 illustrates that predictions made using this model can be used to determine the experimental limit loads very accurately, for sandwich samples loaded at the neutral axis. The experimental error due to measurement uncertainties can add as much as five percent to the difference between theoretical and measured values presented in Table 6.

TABLE 6. UNBALANCED, SANDWICH CONSTRUCTION BUCKLING AND CORE SHEARING LOADS

	Buckling Load (lbf per unit width)			Post Buckling Load (lbf per unit width)
Length inches	Actual	Theoretical	Percent error	Core Shearing
36	1330	1375	3.3	1124
24	2250	2268	1.0	2051
18	2900	2934	1.2	2818
12	3650	3711	1.6	3650

C. POST BUCKLING PHENOMENA

In all cases, core shearing which resulted in deformation of the honeycomb core and face materials occurred. This was a post buckling event except in the 12 inch sample which appeared to fail catastrophically in shear and buckling simultaneously. The deformed regions occurred randomly at either end of the specimens, near the end supports. Small transverse wrinkles and crackling noise preceded the rapid deformation and load loss. An illustration of the resulting "S" bend shape of the sandwich faces with core crimping is shown in Figure 19. The loads at which core shearing was observed have been previously recorded in Table 6.

The occurance of the shear crimping failure at the ends of the specimens can be understood by relating the transverse deformation to the slope of the displacement which occurs.

The buckling modes are sine functions, and therefore, the slope, or derivative of the sine function, is a cosine function. A maximum value of the slope, and hence the shear deformation occurs at the ends of the specimens.

Four unbalanced, sandwich composite samples loaded eccetrically, approximately .15 inches off the neutral axis, failed in the same manner previously described, except that the failure loads were about 20% lower in the off axis tests (Table 7). The 18 inch sample failed by shear crimping quickly after additional load was applied to the buckled configuration.

When the specimens were unloaded and removed from the test fixture, the materials proved quite resilient. The titanium retained a small amount of deformation, and few wrinkle lines were left on the core material, where core shear failure had occurred. Plots of load versus strain, displacement and deflection for a sandwich composite sample are presented in Figures 20-24.

TABLE 7. UNBALANCED, SANDWICH CONSTRUCTION BUCKLING AND CORE SHEARING LOADS, ECCENTRICALLY LOADED

Buckling Load (pounds force per unit width)		Post Buckling Load (pounds force per unit width)
Length inches	Actual	Core Shearing
36	1106, 1178, 1051	986, 1156, 1033
18	2273	2273

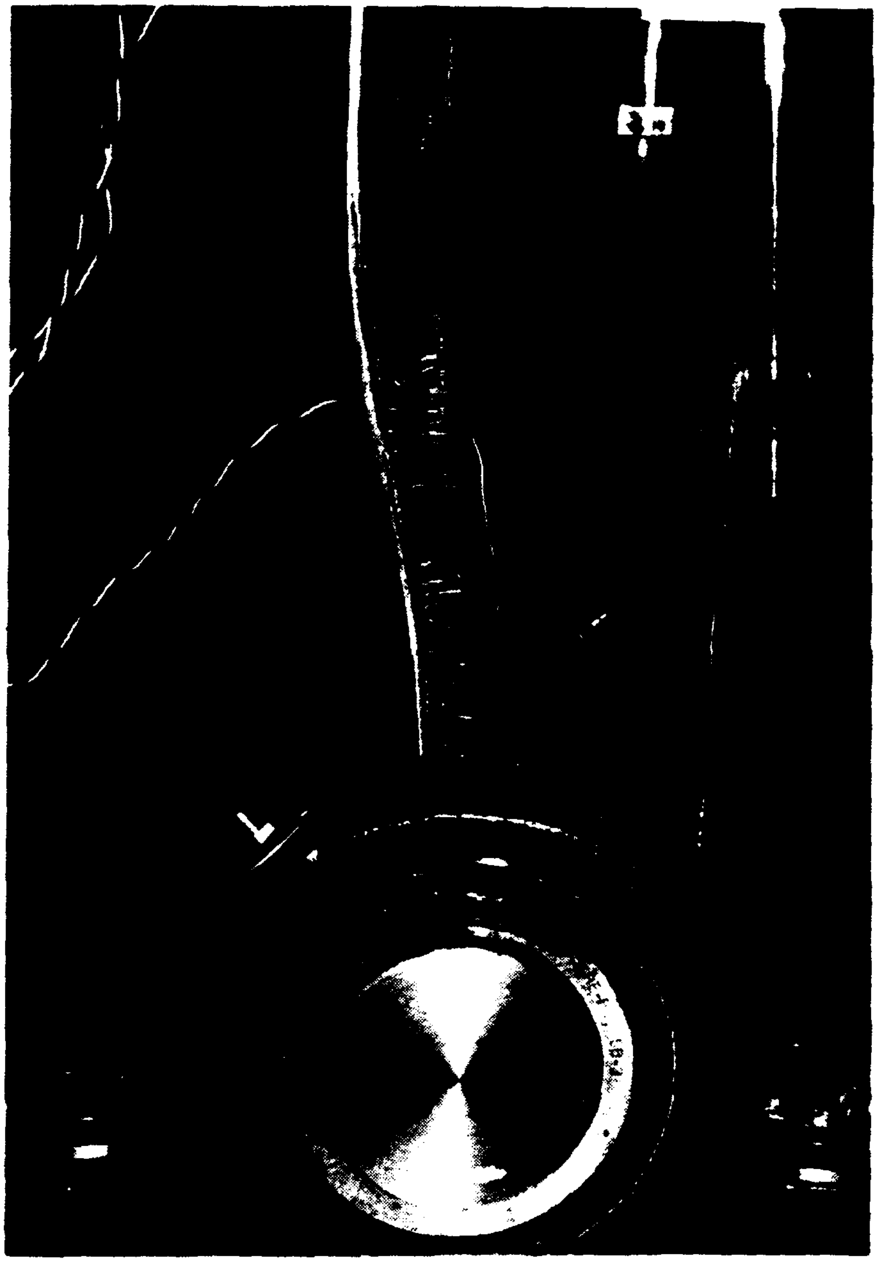

Figure 19. Post Buckling Core Shearing with Face Material Deformation.

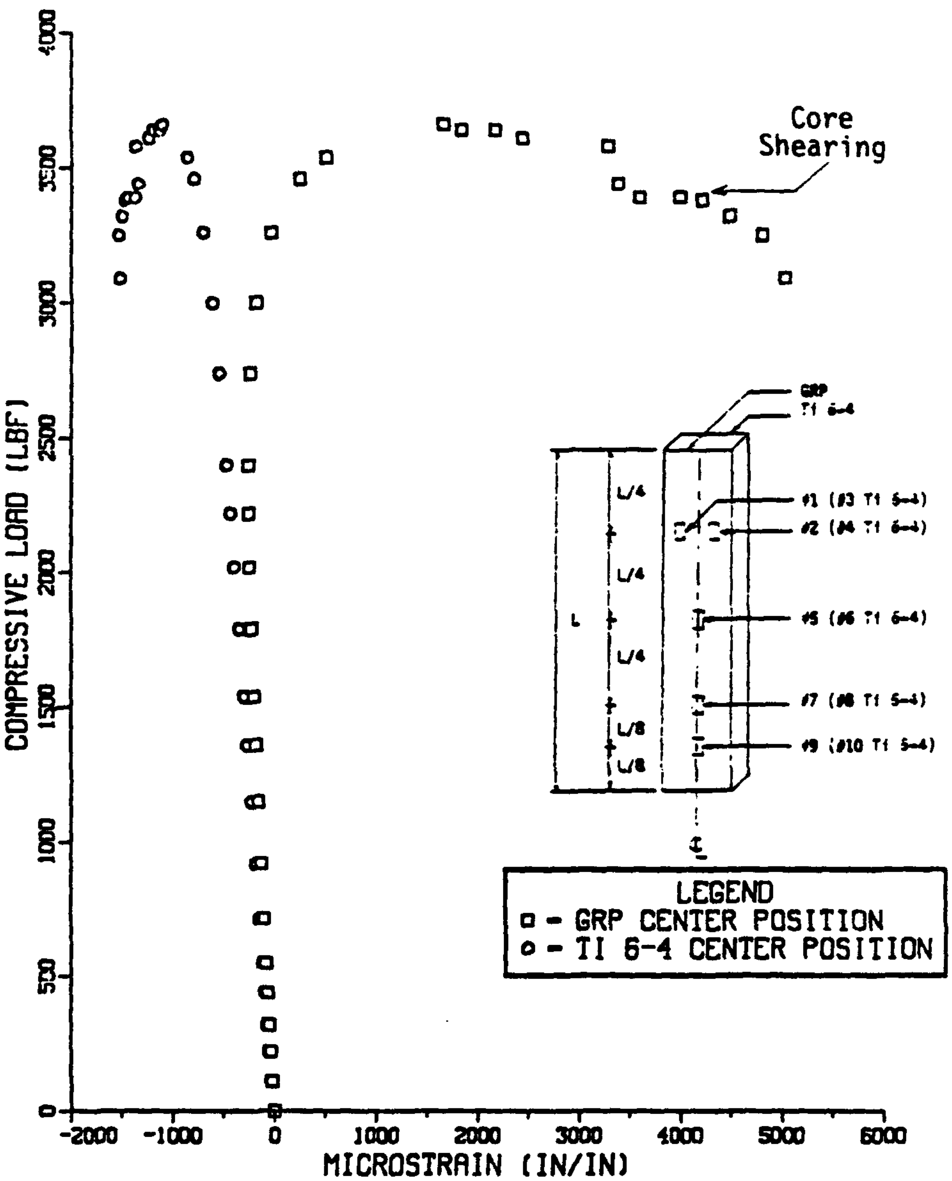

Figure 20. Load Versus Strain, Unbalanced Sandwich Composite.

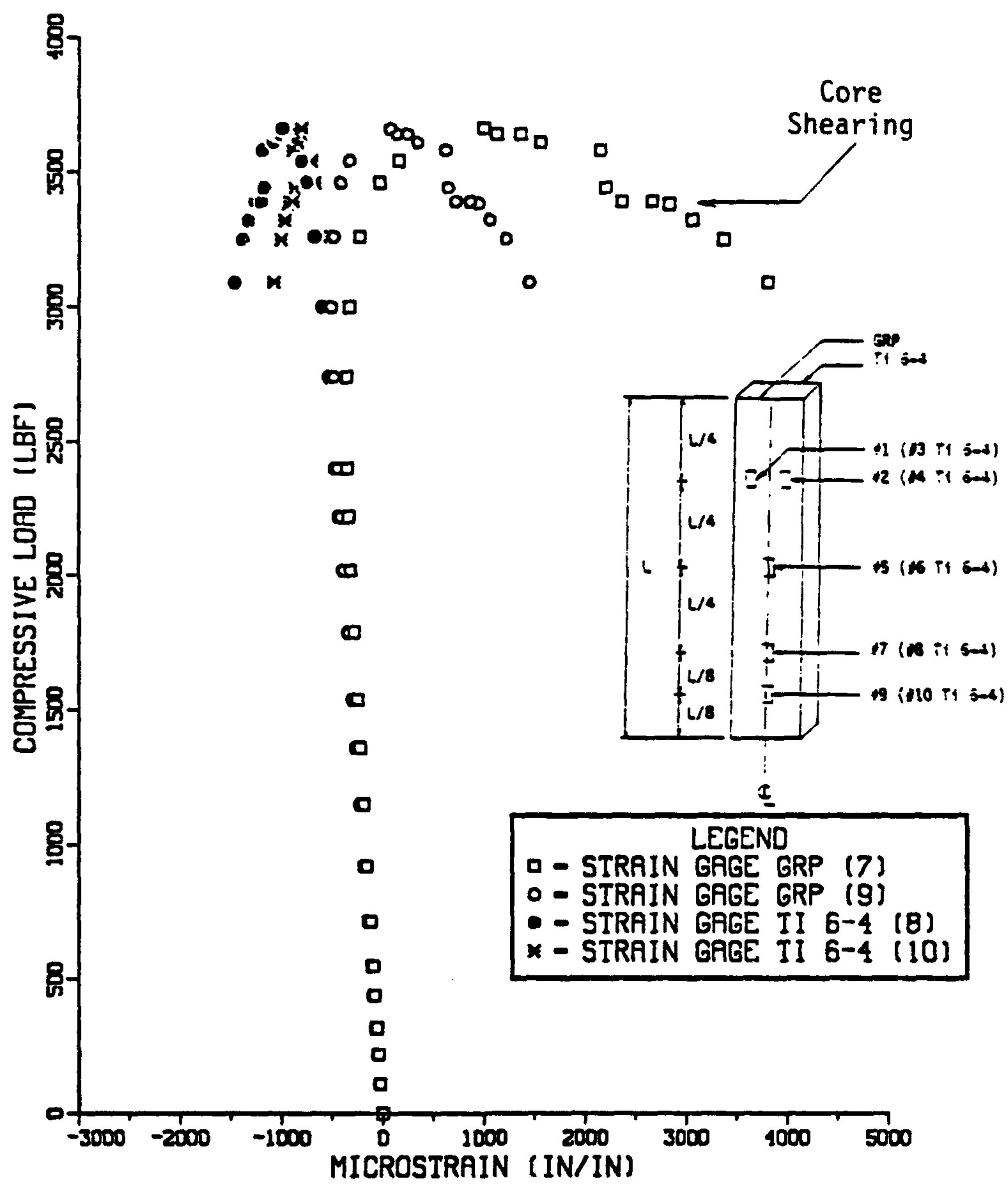

Figure 21. Load Versus Strain, Unbalanced Sandwich Composite.

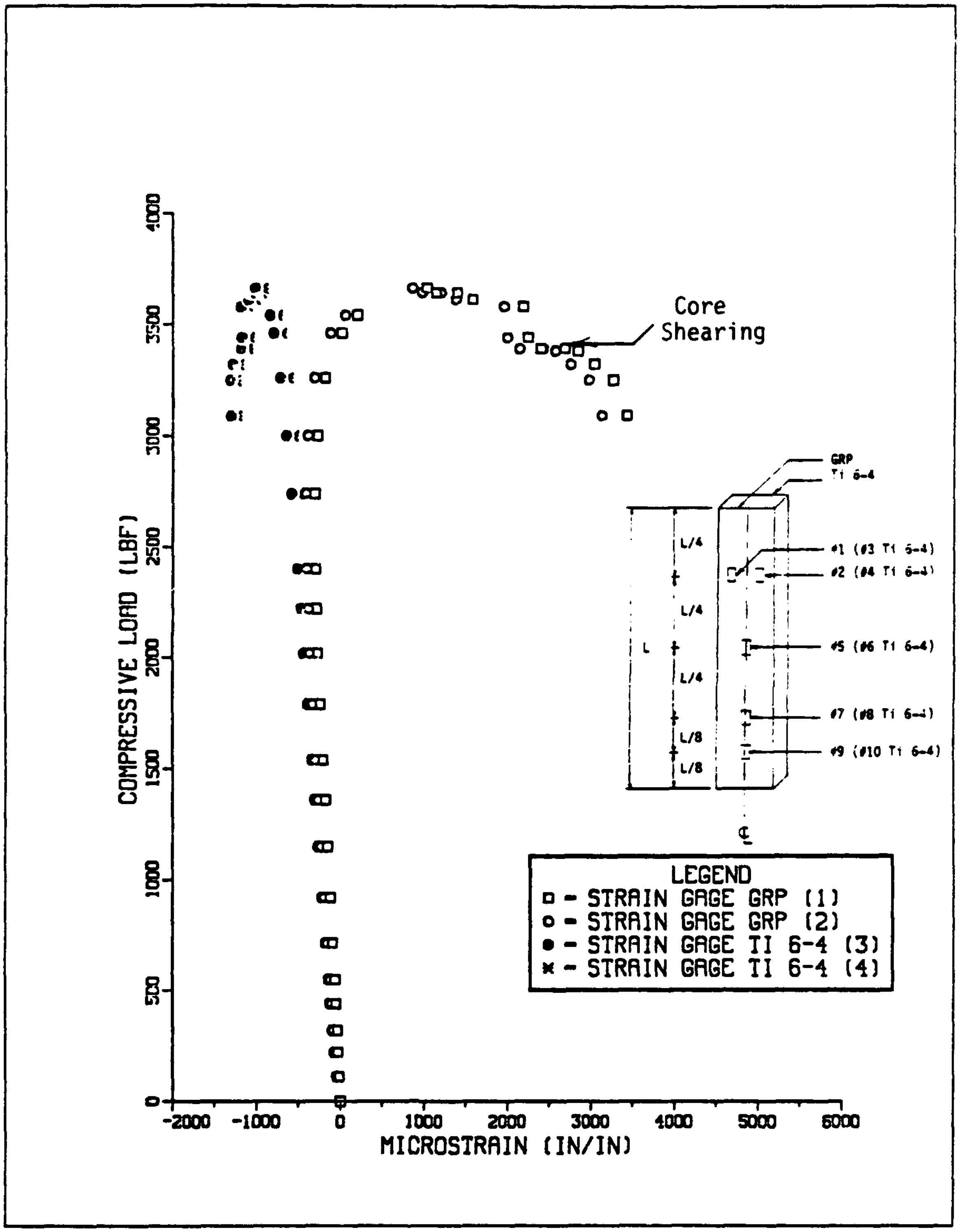

Figure 22. Load Versus Strain, Unbalanced Sandwich Composite.

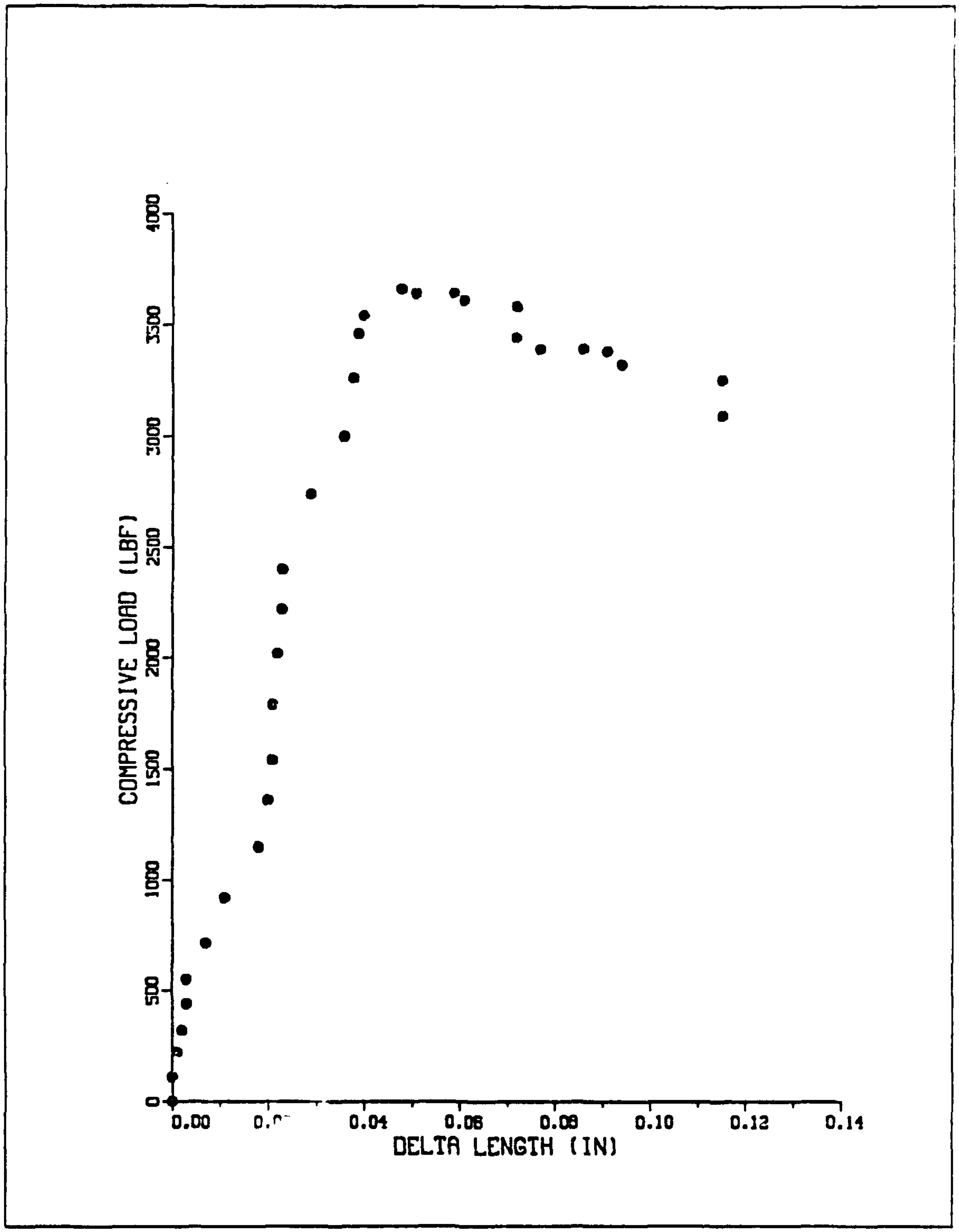

Figure 23. Load Versus Axial Displacement, Unbalanced Sandwich Composite.

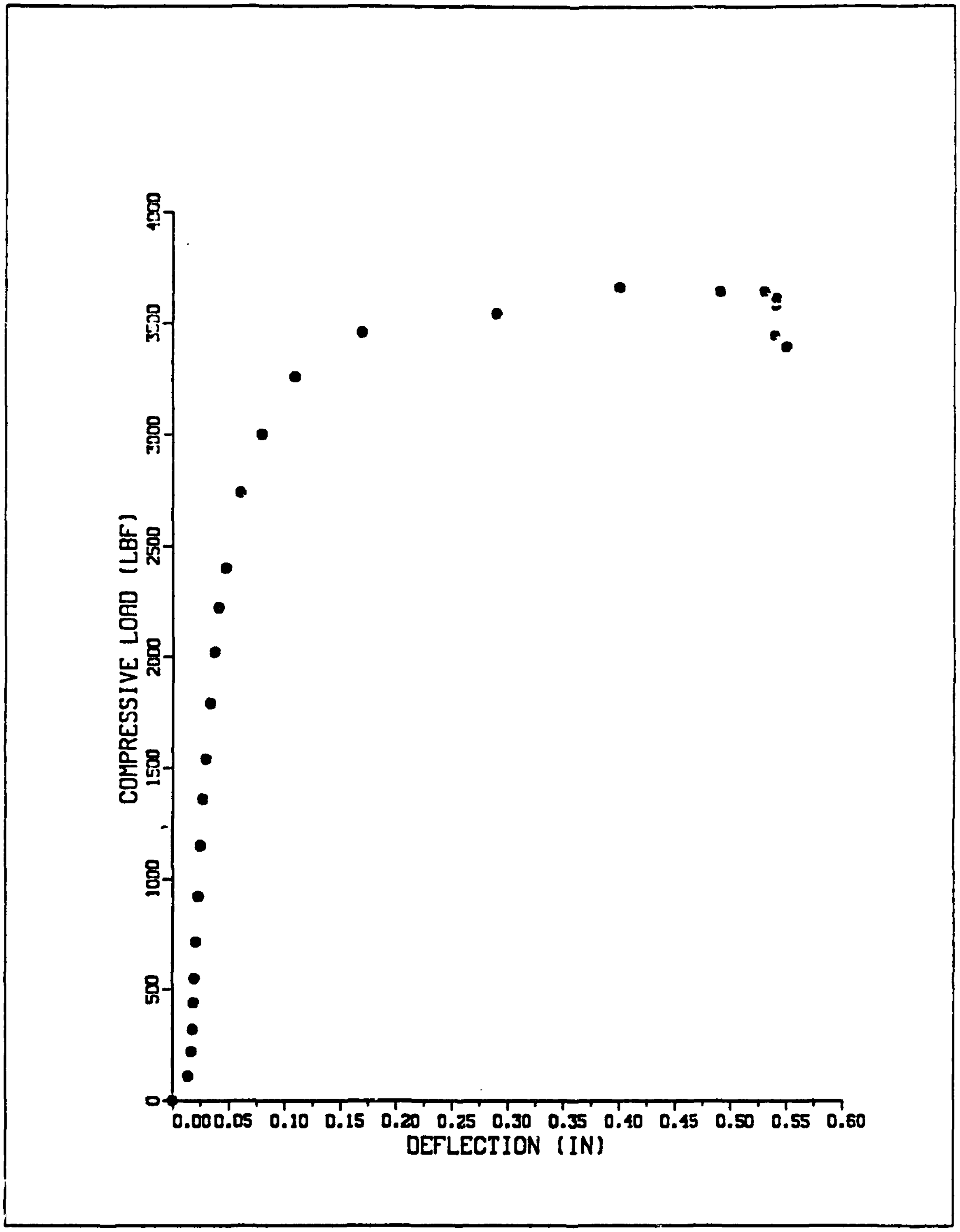

Figure 24. Load Versus Center Deflection, Unbalanced Sandwich Composite.

VI. SUMMARY OF TEST RESULTS

The limit failure loads for titanium 6Al-4V and woven glass in vinyl ester face materials tested by this thesis work were predicted by Eulerian theory accurately. The Ti6-4 skin material was found to be about ten times as stiff as the GRP.

The designed experimental test fixture was able to accomplish ideal simply supported loading conditions.

Two modes of failure were observed in the unbalanced, sandwich composite studied. The primary mode of failure in the sandwich constructions was overall buckling. Subsequent to overall buckling, core shear crimping resulted when an increase in loading was attempted. The core shearing and "S" shaped bending occurred randomly, at either end of the test samples.

In order to accurately predict the limit failure loads for the unbalanced, sandwich composite configuration, the shearing deformation energy of the honeycomb core had to be included. This factor was more influential as the length of the specimens became shorter. Figure 25 illustrates the experimental results with the two theoretical predictions.

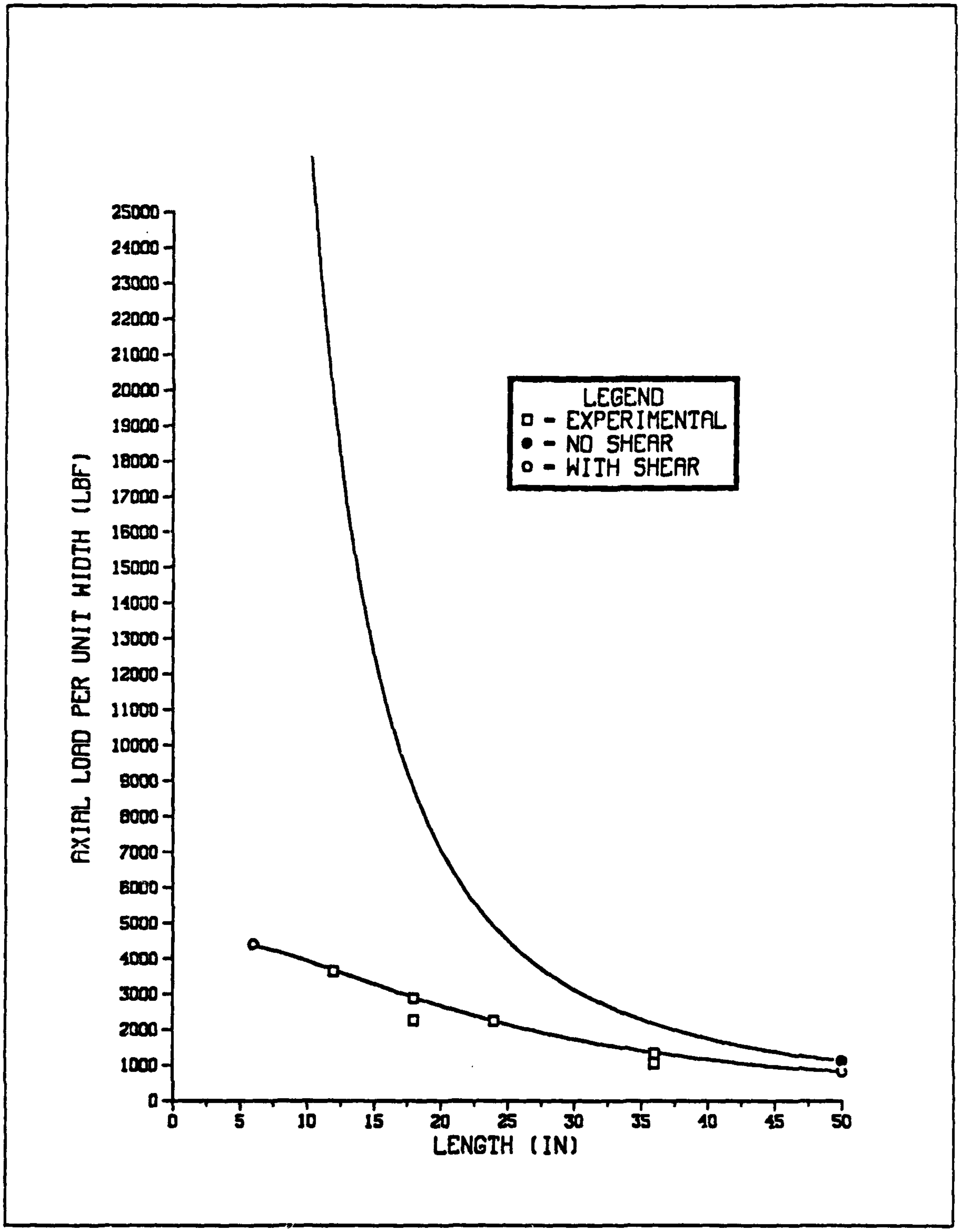

Figure 25. Two Predictions of Axial Load Per Unit Width with and Without Shear Energy Deformation, and Experimental Results.

VII. CONCLUSIONS

Both theoretical and experimental analyses have been given for the buckling instability of an unbalanced, honeycomb sandwich composite beam with woven glass vinyl ester and titanium 6Al-4V faces. The theoretical analysis takes into account the bending strain energy of skin materials as well as core material shearing energy.

Uniaxial compressive load experiments were conducted on eight unbalanced, sandwich composite beams simply supported on the loaded ends and the overall limit loads observed were within 10% of the theoretical predictions using the analytical model developed. Core shearing initiated quickly after load was applied to a buckled specimen, and the specimen abruptly lost its load carrying capacity. Other possible modes of failure in sandwich structures were not observed.

The ideal condition of simple support was very closely achieved by the described experimental design. No local damage was introduced by the end supports.

It is recommended that further experimental work be conducted to analyze the effects of varying face and core material thicknesses, as well as aspect ratios, to better predict the bucking instability of the APMS prototype.

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